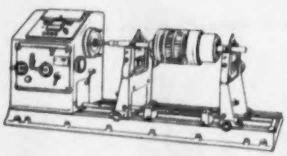
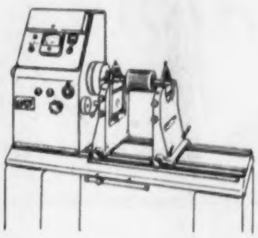
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

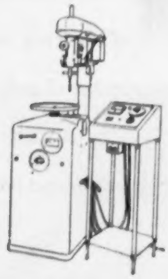
Vol. 45 No. 10 OCTOBER 1955 PRICE: 3s. 6d.



Model UA. for components from 2.2 lbs. to 22,000 lbs.



Model UA. 10 and 30 for components from 0.22 lbs. to 66 lbs.



Vertical Single Plane Balancing Machine, Type SAE.40, and Drilling Attachment.

IF YOU WOULD LIKE ADVICE ON YOUR NEXT BALANCING JOB, SEND US A COMPONENT DRAWING FOR EXAMINATION

Model DU. for components from 0·08 lbs. to 11 lbs.

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DYNAMIC BALANCING

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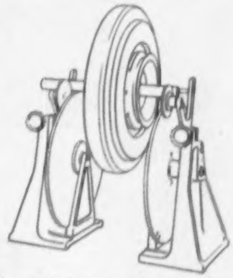

Static Balancing Machine, Disc-type SE.

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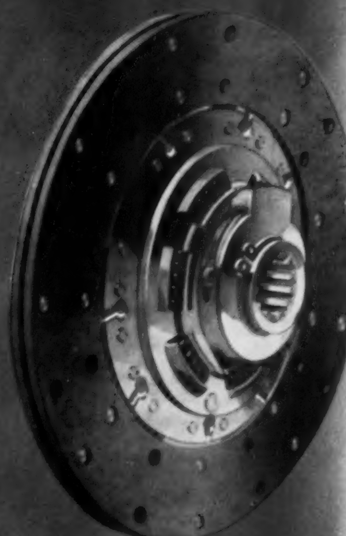

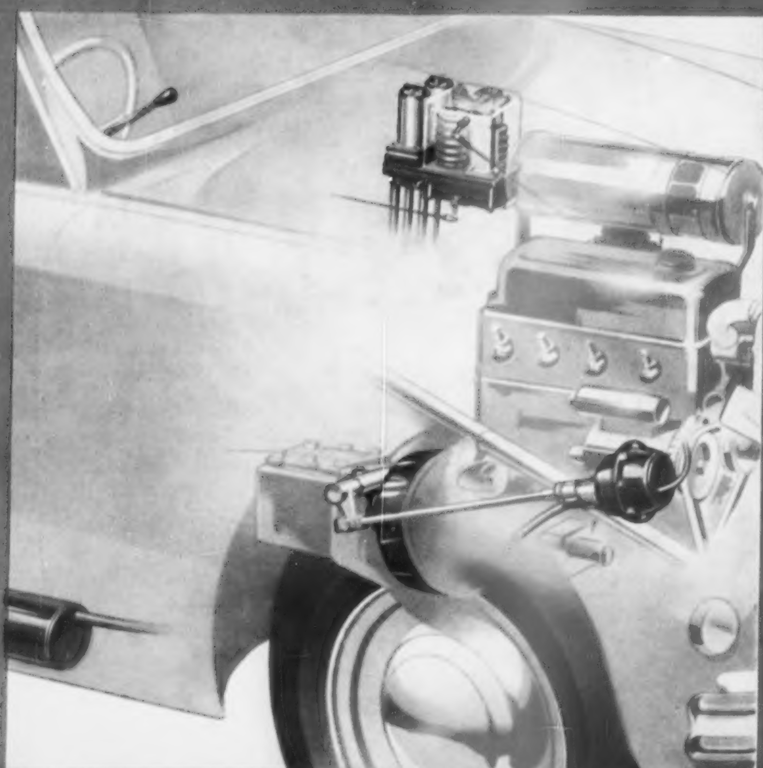

The clutch is based upon the standard Borg & Beck, developed as a smooth-acting centrifugal clutch. When the accelerator is pressed to speed up the engine the Manumatic clutch engages automatically and the car moves smoothly away. An electrical contact device is embodied which — if the engine speed falls below the driven-plate speed, causes a small vacuum servo to open the throttle.

The act of moving the gear lever makes a contact which sets the clutch-actuating servo in action; this withdraws the clutch and also ensures its smooth re-engagement.

The units are simple and easily installed — provision is made to ensure freedom from 'hunting' and to permit a 'tow' start.
On the road, however crude the operation by the driver may be, the action is equal to the finest performance put up by a skilled demonstrator.

THE *Manumatic*

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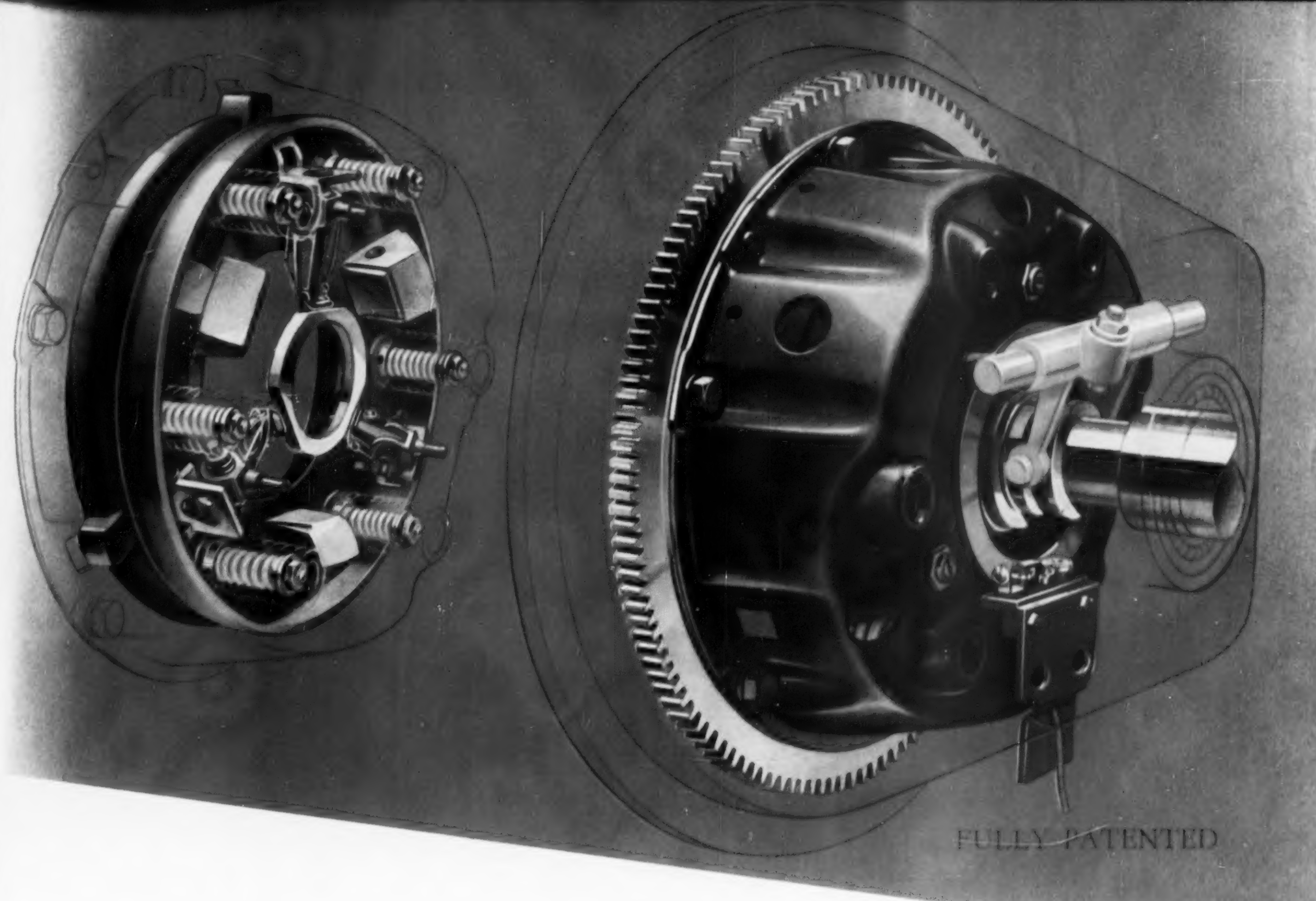

The MANUMATIC transmission system has been especially developed to suit the requirements of the ordinary British car; providing an efficient transmission system for two-pedal control and yet without needing any major alterations in the existing design.

The system is fundamentally simple and effective; it requires only a minimum of maintenance.

It is a logical development by the foremost transmission engineers in this country, Borg & Beck Company Ltd., the largest British manufacturers of frictional clutches for automobiles

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For all water pump and similar applications.

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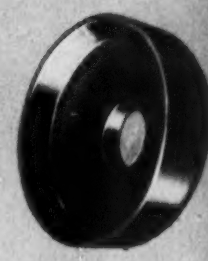
For all types of hydraulic and pneumatic applications.



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★ *unequalled radial load capacity*

★ *low unit cost*

★ *long service life*

★ *compactness and light weight*

★ *runs directly on hardened shafts*

★ *permits use of larger and stiffer shafts*

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Illustration left: A standard 5kW unit, for brazing carbide tool tips; photograph by courtesy of EDIBRAC LTD., Broadheath, Cheshire, manufacturers of Tungsten Carbide products.

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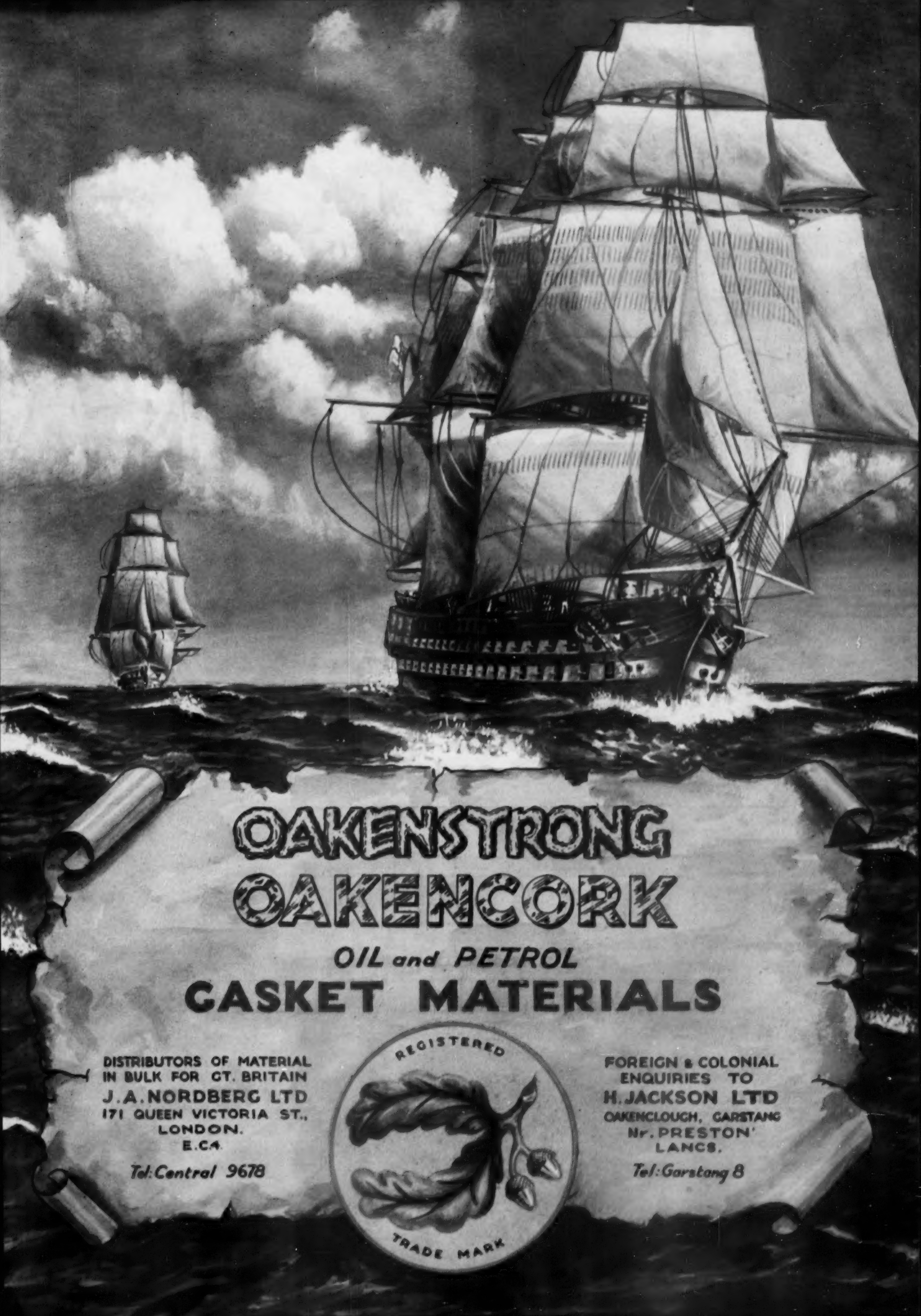

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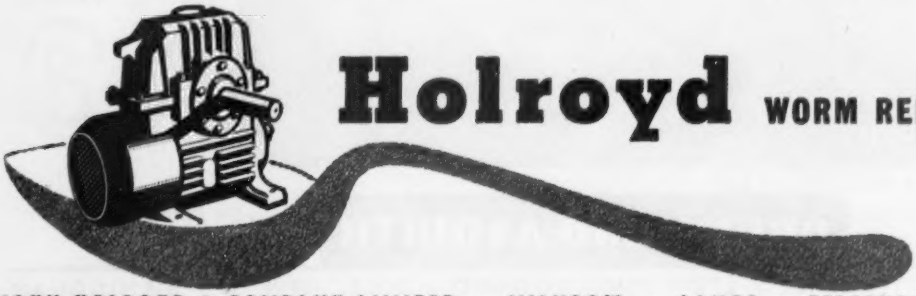

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helping to break production records in the Motor Industry

Maximatic Multi-Tool Production Lathes are to be found in their hundreds in Britain's automobile factories. The output record of the industry is sufficient testimony to their capacity for continuous, fast, accurate production.

Maximatics are simple to operate and in many cases, as in the example below, one operator can easily take charge of three machines.

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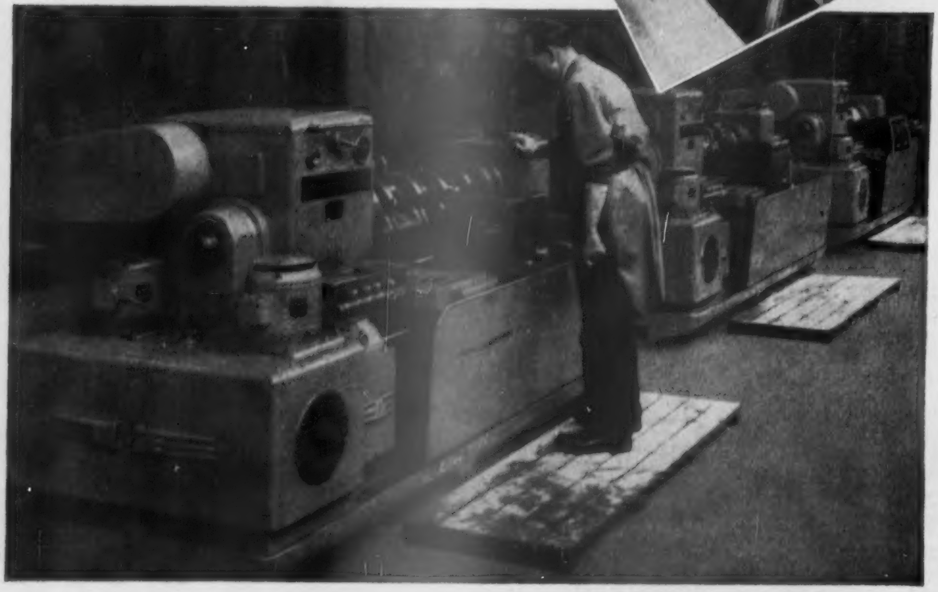

Drummond Maximatics turning crankshaft journals and pins. Photographs by ccurtesy of Morris Motors Ltd., Engines Factory, Coventry

Sales and Service for . . . **DRUMMOND-ASQUITH** *. . . the British Isles*

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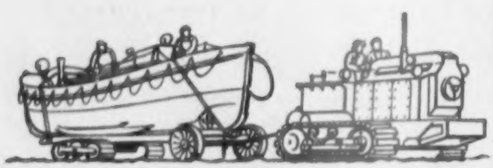
Shell Rotella Oil, one of the well-known world-wide Shell Seashell Brands of lubricating oil, gave complete protection to a diesel engine submerged in sea water for seven days.

A number of special crawler tractors of very large size, powered by a Meadows diesel engine of 95 b.h.p. normal rating, the weight of the machines being 30,000 lb., have been built for the Royal National Lifeboat Institution for towing lifeboats into and out of the sea.



These tractors are manufactured by John Fowler & Co. (Leeds) Ltd. They are required to be operationally watertight to a depth of 5 ft., and capable of being sealed immediately for complete submersion if required due to unexpectedly severe conditions. The manufacturers are making them watertight to 8 ft. and, following tests with a prototype, have arranged for instantaneous sealing of the housing by mechanical means.

During one test the entire engine unit — whilst still warm from running—was, by submersion, filled with sea water and remained in this condition for a period of seven days before being opened up and dismantled for complete examination. ALL THE WORKING PARTS WERE IN PERFECT CONDITION and no damage whatsoever from sea water had been suffered, apart from two spots of rust approximately $\frac{1}{8}$ in. diameter on a bearing cap for the rocker shaft and on a web of the crankshaft.

Shell Rotella Oil had given complete protection from corrosion.

Shell Rotella Oil

Shell Rotella Oil is approved for automotive and other lubrication applications by The Marshall Organisation (of which John Fowler & Co. is a member company), Henry Meadows Limited, and the leading diesel engine builders in this country.

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were combined on 1st October, 1955 in order to provide a thoroughly comprehensive and effective service to the welding industry.

The headquarters of the new organisation—named Quasi-Arc Limited—are at Bilston, Staffordshire, with works at Bilston and Gateshead.

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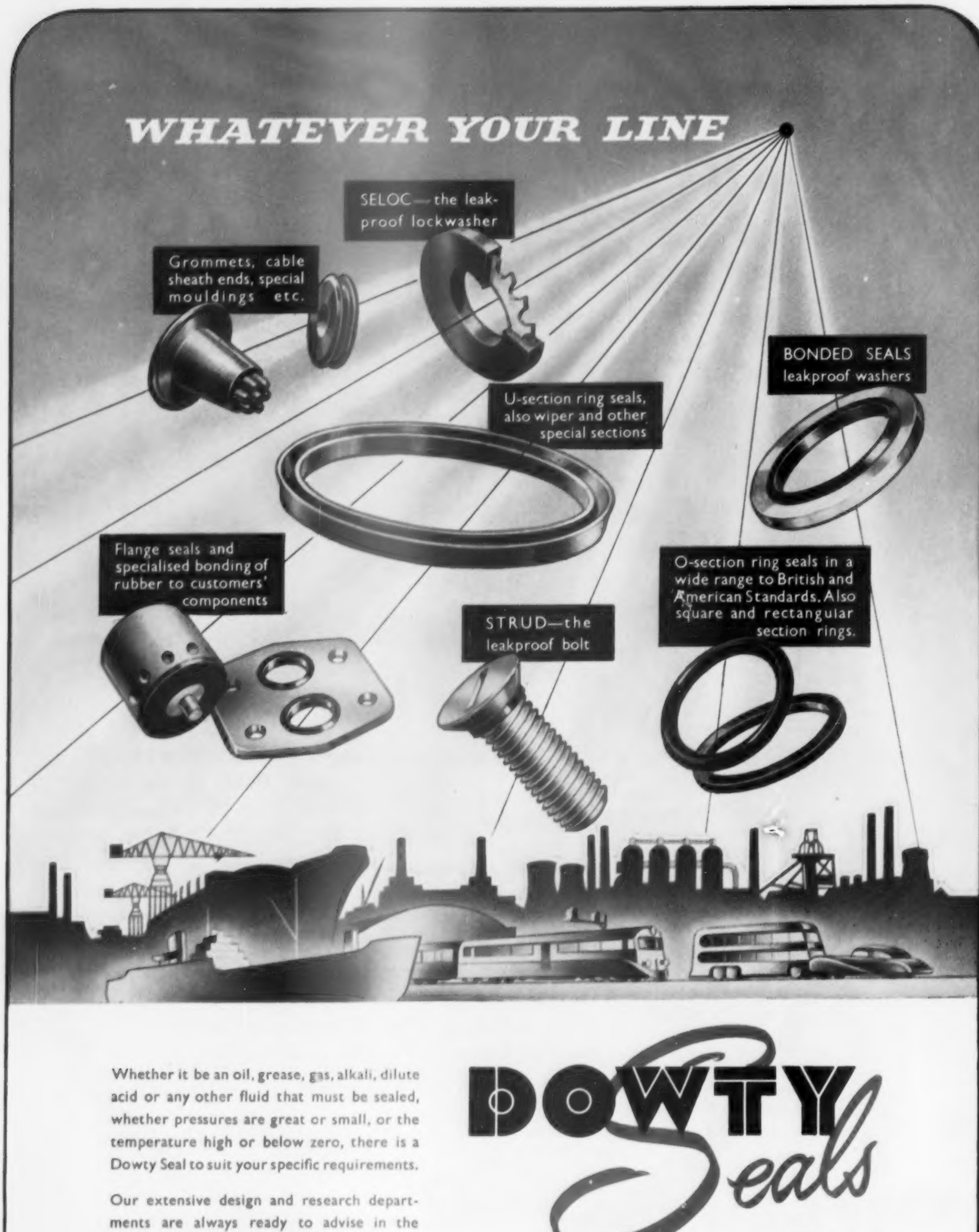
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SELOC—the leak-proof lockwasher
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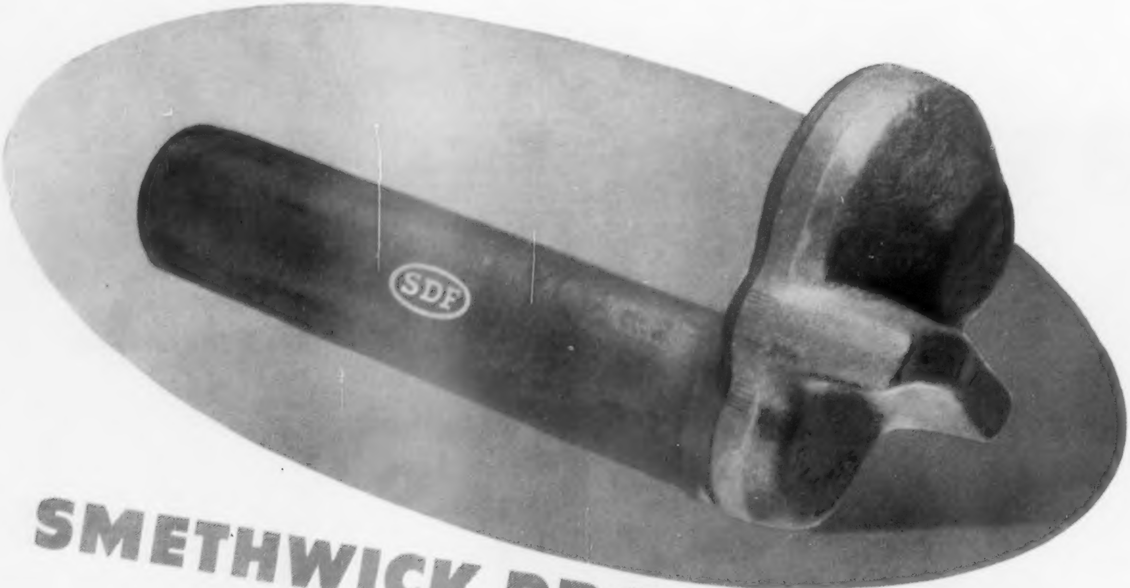
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Valve Master Refacer

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The B&D Valve Shop *incorporates the Valve Master Refacer, the Vibro-Centric kit and the Carbon Removing kit in a transportable cabinet.* ***The 55° Vibro-Centric*** *Valve Seat Grinder used for working in confined spaces.*

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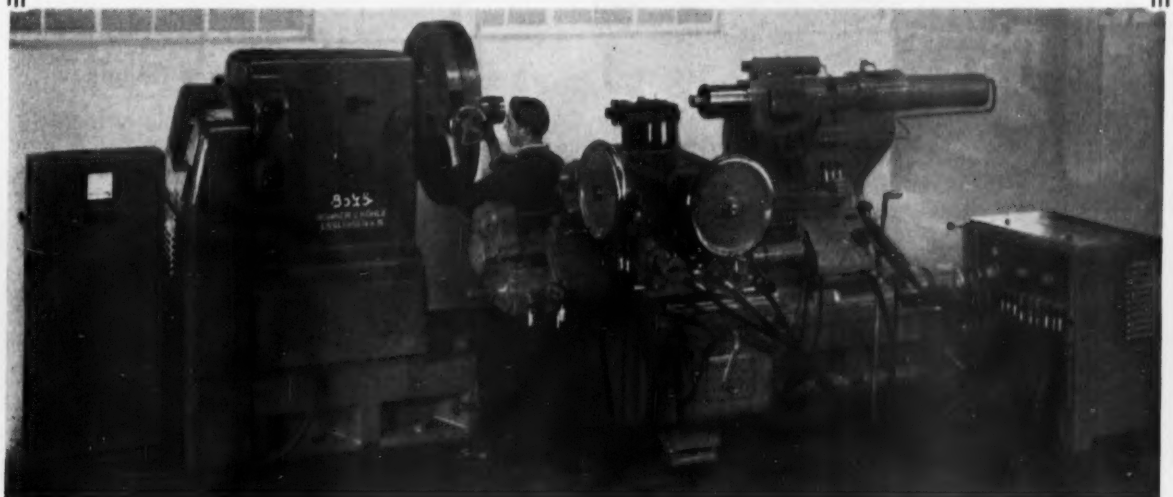

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* For heavy duty Spinning of Stainless Steel, aluminium, etc.
* May also be used for Cold Forming operations.
* Swings a maximum of 86in.
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* Supplied with front slide only or front and back slides as required.
* Hydraulic copying attachment indispensable for complicated work.

Our representative will be pleased to call and discuss this machine with you in detail.

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September, 1955.

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Coming at a time when price rises are a daily expectation, you will be pleased to learn that the effects of our new plans for manufacturing cutters on a larger scale than hitherto, with a degree of rationalisation in design and manufacture, are quicker deliveries and lower production costs. And these lower costs are now to be passed to you in the form of lower prices.

The new prices are shown on the last page of this inset. Compare them with what you are now paying, and you must agree that cutters of this quality have never been available at such attractive prices. Compare, too, the price of replacement blades and the simple method of blade renewal and reservicing.

It is possible you are considering replacement carbide cutters at this moment. If you are not already familiar with Wickman quality and performance, this is your opportunity to experience it.

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Yours faithfully,
for WICKMAN LIMITED,

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Sales Manager, Wimet Division.

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Whether you buy on price, delivery, or performance, this range of Wickman Milling Cutters has everything you've ever wanted.
★ Inserted blades in heavy forged steel bodies.
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★ Readily available replacement blades at low prices.
★ Standard cutters available on quick delivery (many from stock).
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If you are not already using carbide face mills, here's your opportunity to experience high performance at moderate cost. They will give you quicker floor-to-floor times, more jobs per regrind, infinitely better finish, and lower tool cost per component.
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FOR CAST IRONS
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8 in. DIA.
20 BLADES
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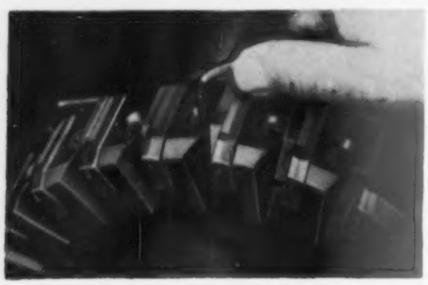
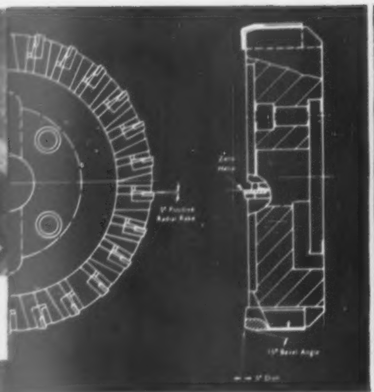
BLADE *Release socket head screw and partially withdraw.*

RENEWAL *Clamping wedge is withdrawn with screw.*

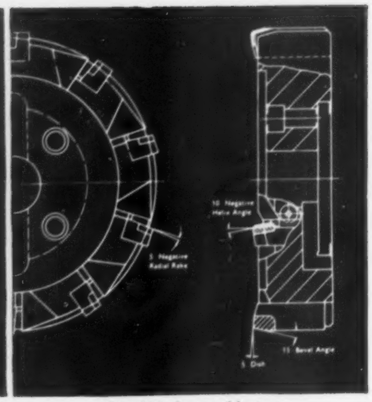
IS SO *Old blade is withdrawn and replaced by new.*

SIMPLE *New blade positioned by setting block; wedge locked by screw.*





The Wickman Ferromill



The Wickman Hi-ten

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FERROMILL — FACE MILLING CUTTER FOR CAST IRON AND NON-FERROUS METALS

Dia.	Number of Blades	Price Cutter Complete £ s. d.	Price Blades Each s. d.	Cutter Code R & L-Hand	Blade Code R & L-Hand
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6″	14	26 10 0	18 6	F-6R F-6L	FBR FBL
8″	20	38 0 0	18 6	F-8R F-8L	FBR FBL
10″	26	51 0 0	18 6	F-10R F-10L	FBR FBL
12″	32	63 0 0	18 6	F-12R F-12L	FBR FBL

Standard Spindle Nose Fitting

HI-TEN — FACE MILLING CUTTER FOR STEELS

Dia.	Number of Blades	Price Cutter Complete £ s. d.	Price Blades Each £ s. d.	Cutter Code R & L-Hand	Blade Code R & L-Hand
6″	6	18 5 0	1 8 6	HT-6R HT-6L	HBR HBL
8″	8	25 10 0	1 8 6	HT-8R HT-8L	HBR HBL
10″	10	34 0 0	1 8 6	HT-10R HT-10L	HBR HBL

Standard Spindle Nose Fitting

PRICES FOR NON-STANDARD SIZES QUOTED ON REQUEST.

These cutters follow the long-established design features of Wickman Wimet-tipped face milling cutters; robust, heat-treated bodies and firmly clamped inserted blades, easy reservicing, and simple blade replacement when eventually necessary. They are equally suitable for heavy roughing and light finishing cuts.

Their low initial cost and inexpensive blade replacement brings a new economy to carbide milling.

If you are already using carbide cutters, here's an opportunity to cut costs.

If you have not before considered carbide cutters, this is your opportunity to secure a two-fold gain; lower costs *and* higher production.

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ECLIPSE
LOW TUNGSTEN
UB 14
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Photo: courtesy 'Metalworking Production'.

SOLEX GAUGES AT ROLLS-ROYCE

This photograph shows a panel of Solex Air Jet Plug Gauges in use at the Crewe factory of Rolls-Royce Ltd.

Solex Plug Gauges for accurate internal measurement

COMPARATOR PLUG GAUGES

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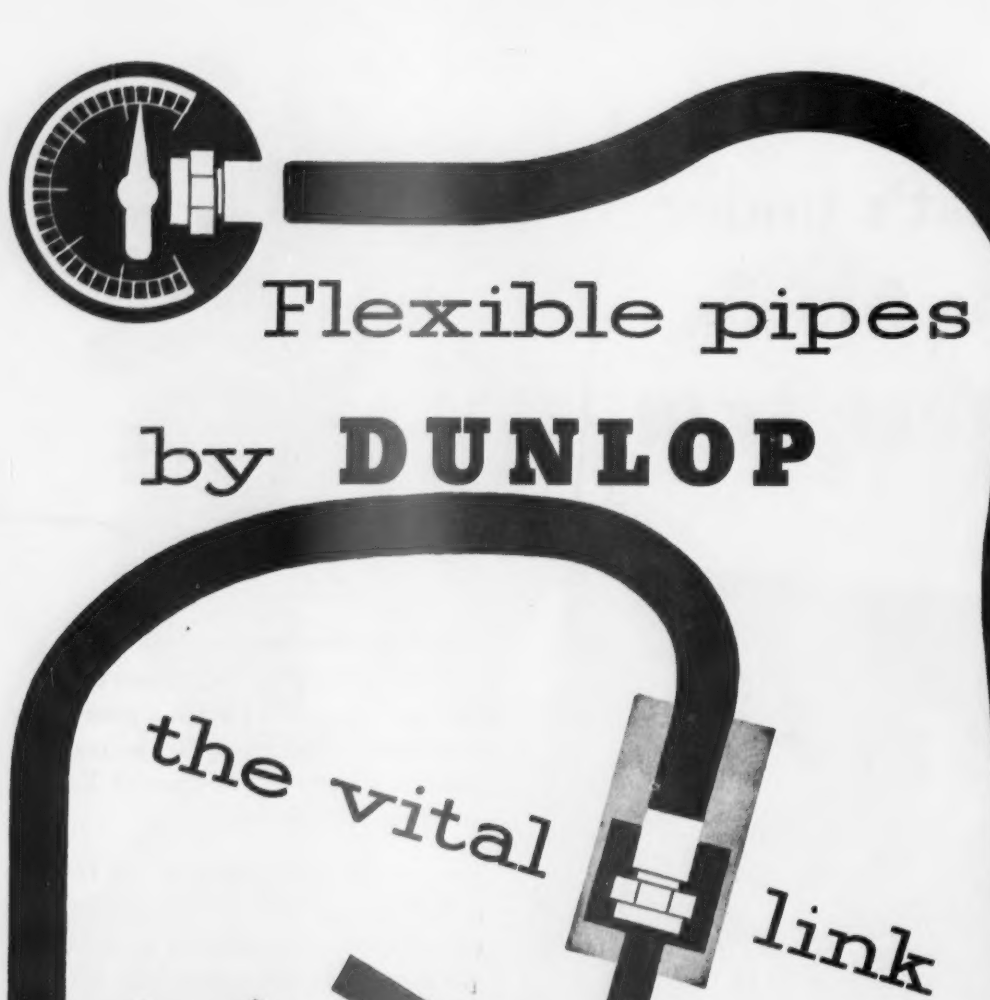
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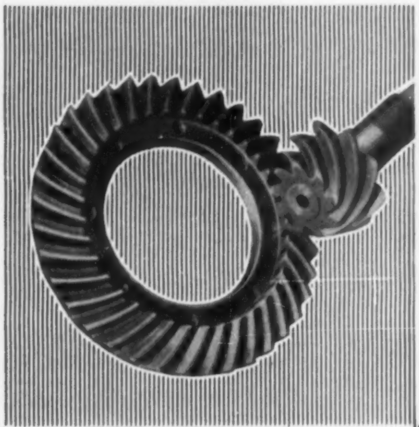
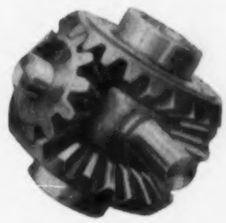
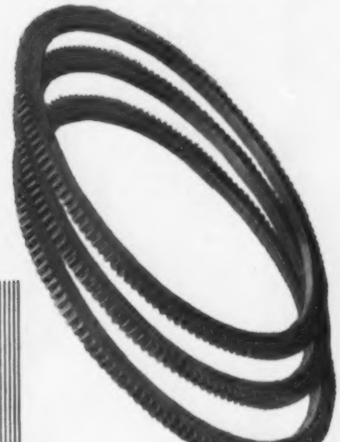

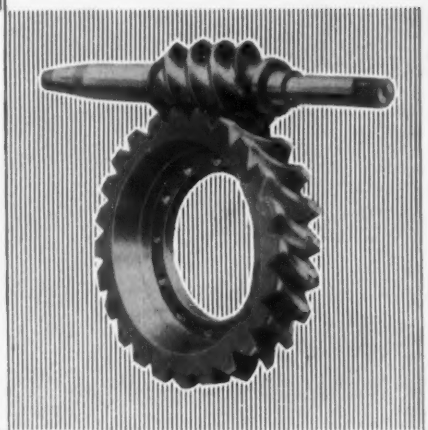

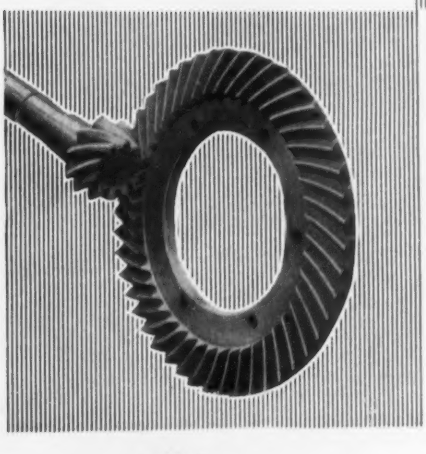

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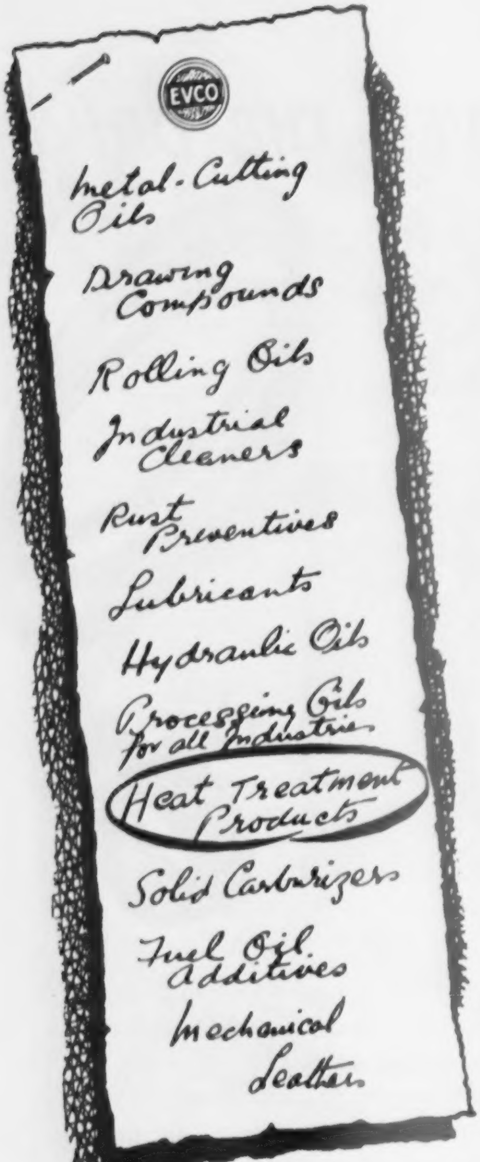

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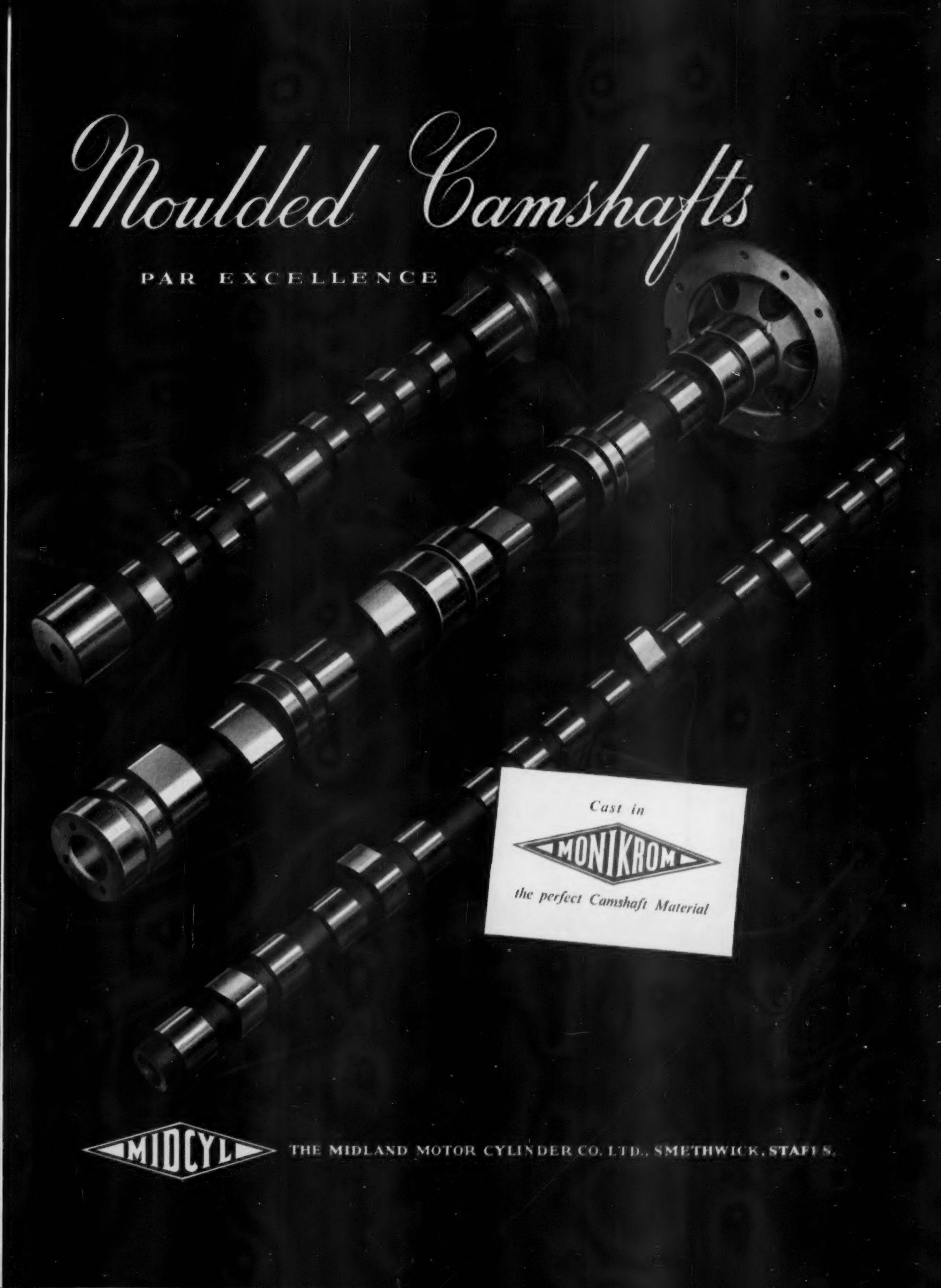
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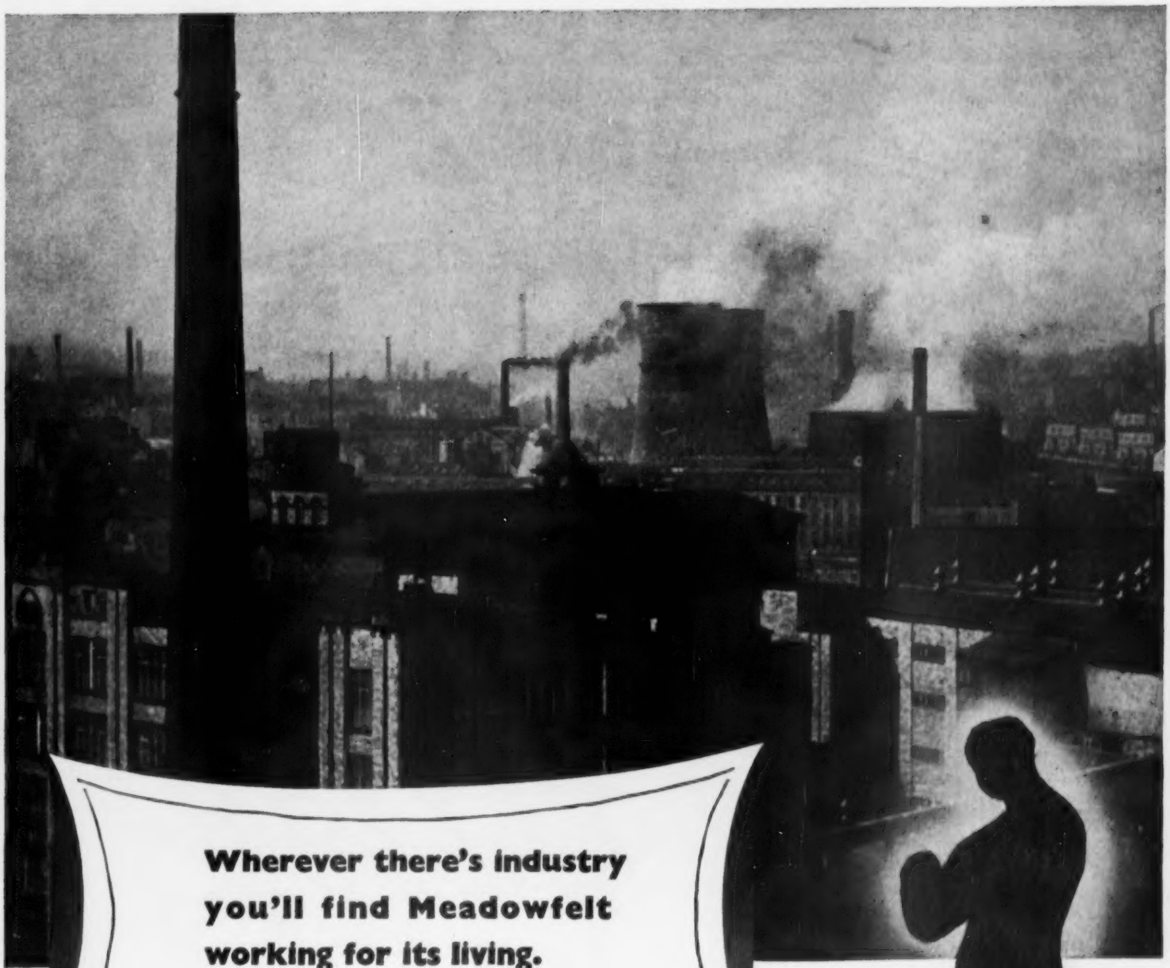
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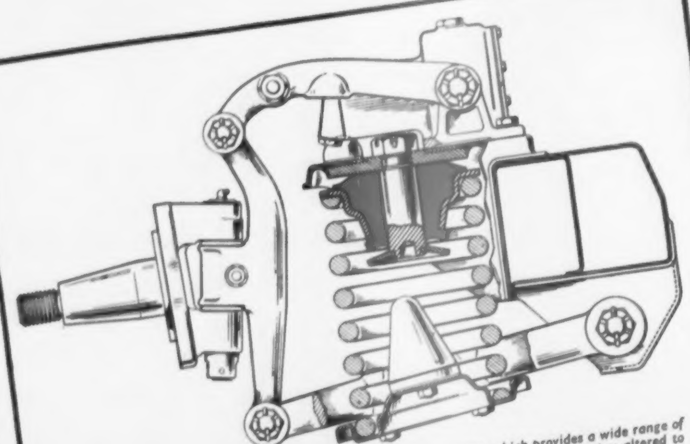
The 'Frustacon' mounting fitted to the I.F.S. of the 1952 Rover '75'. Sectioned to show rubber.



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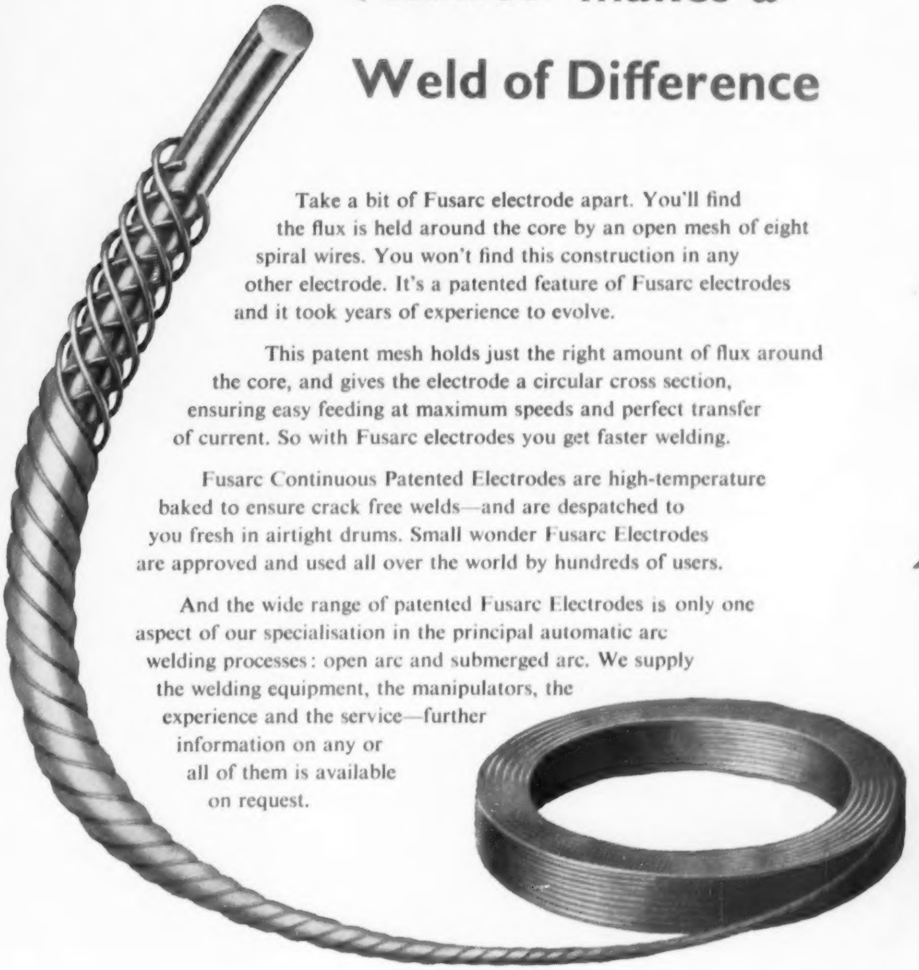

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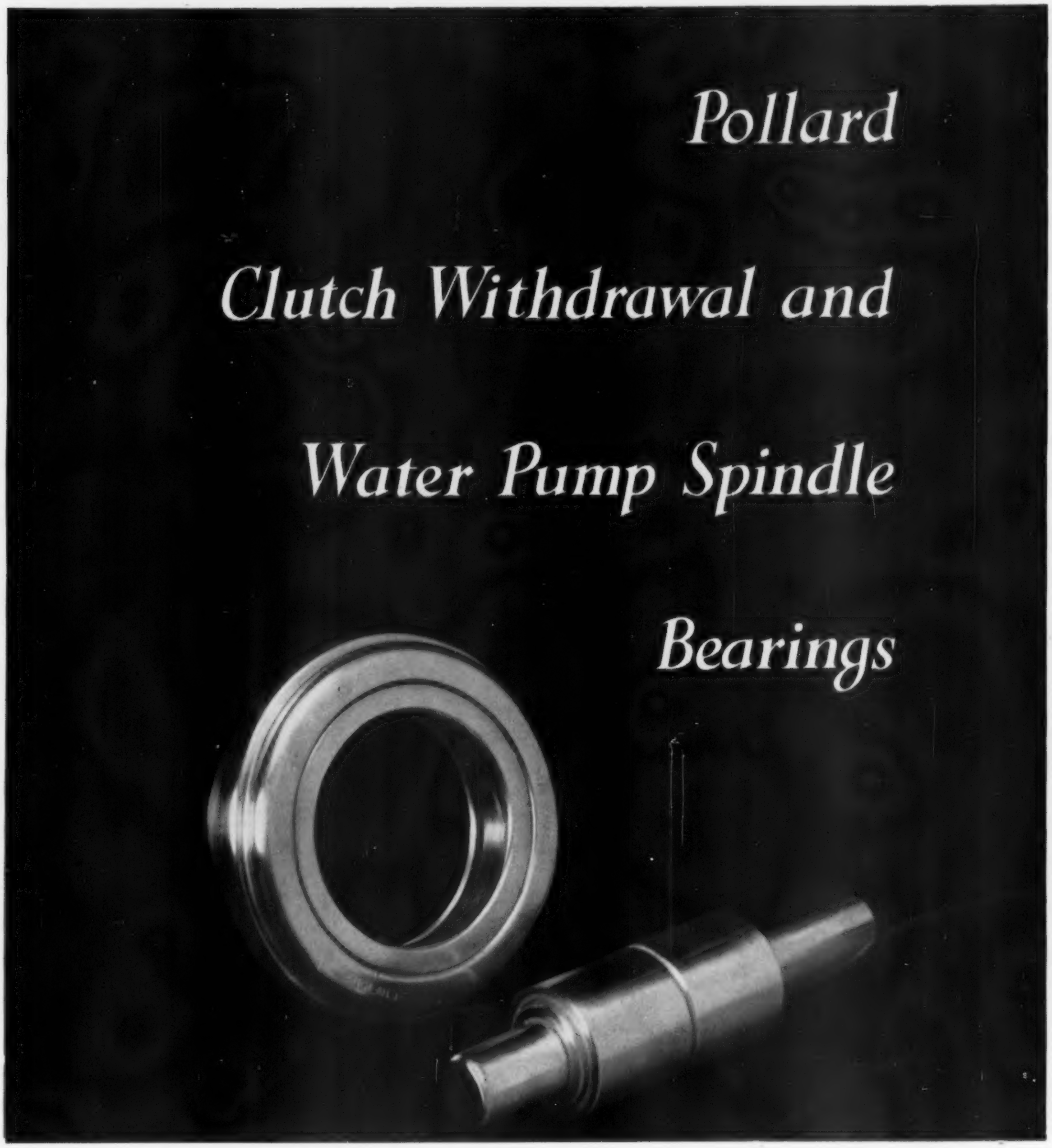
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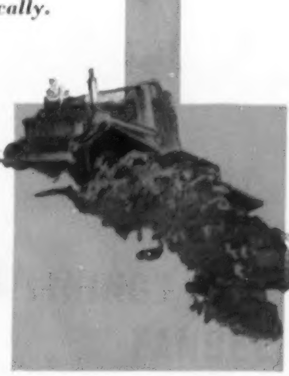

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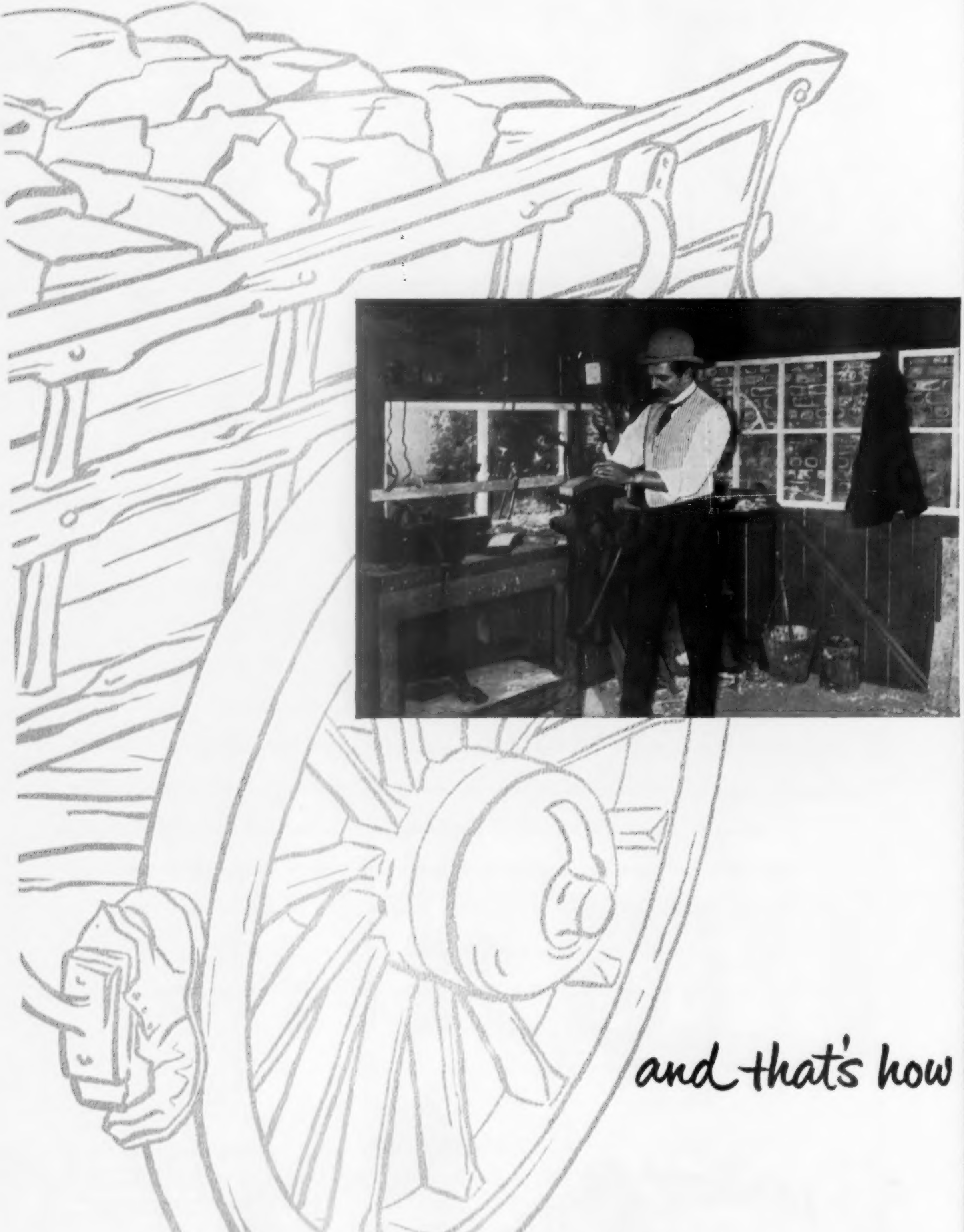

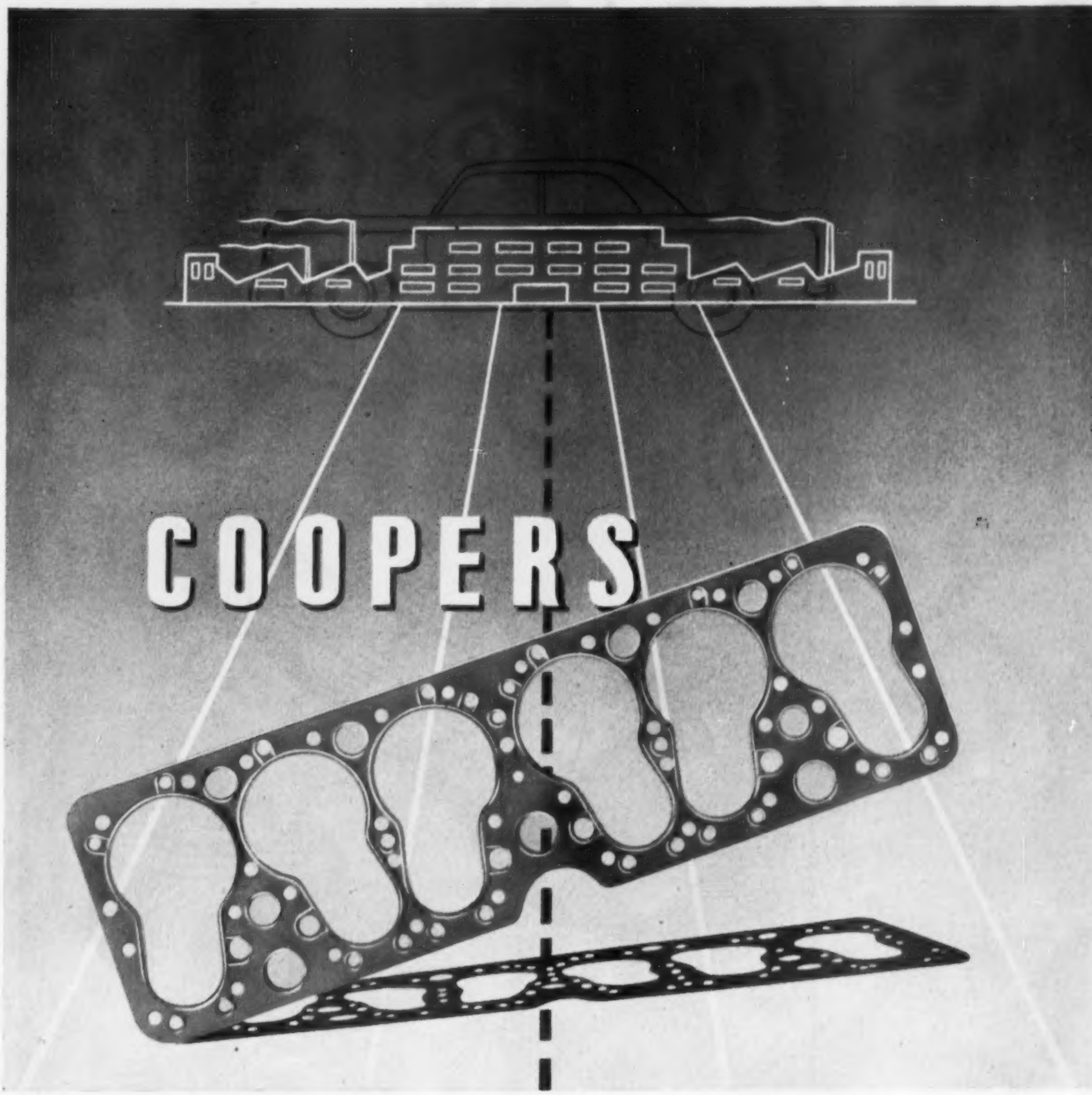
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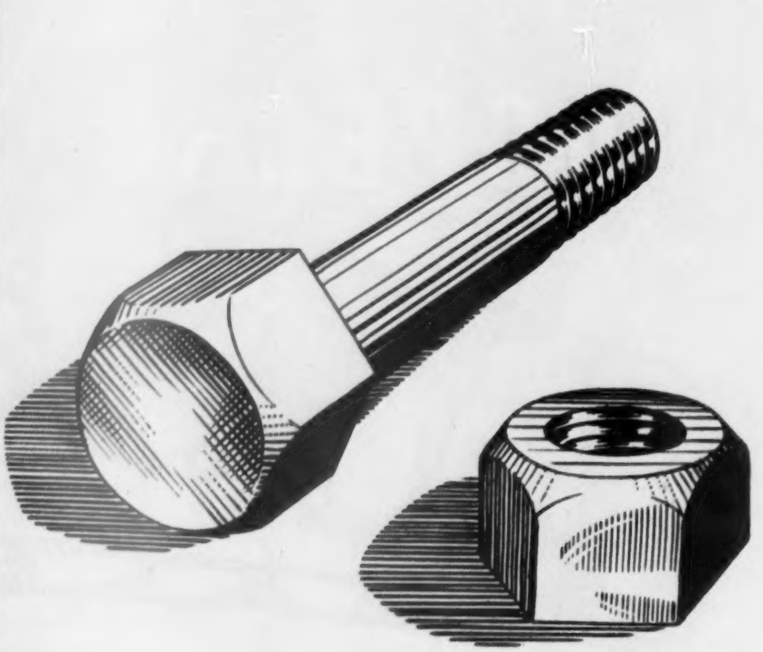

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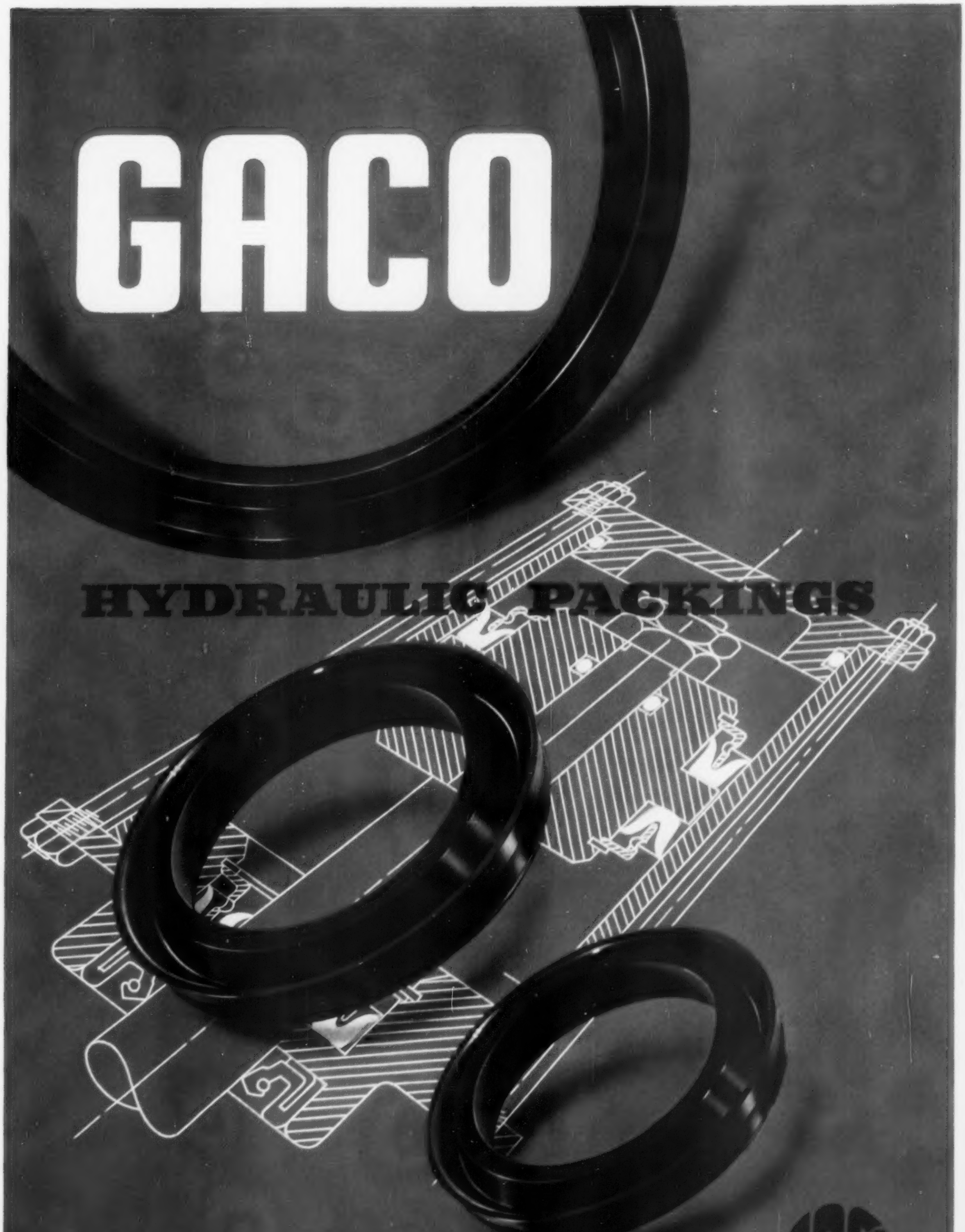
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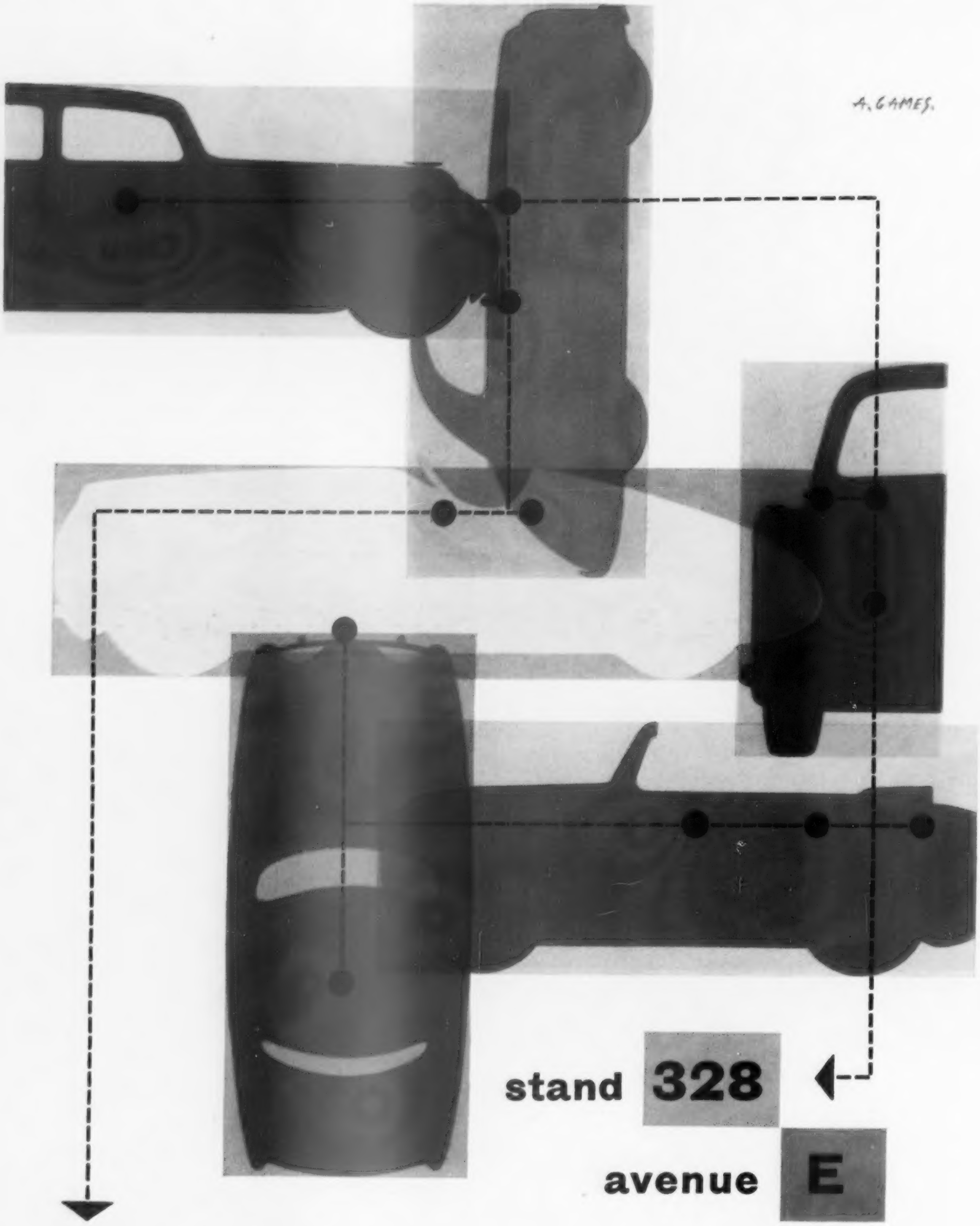
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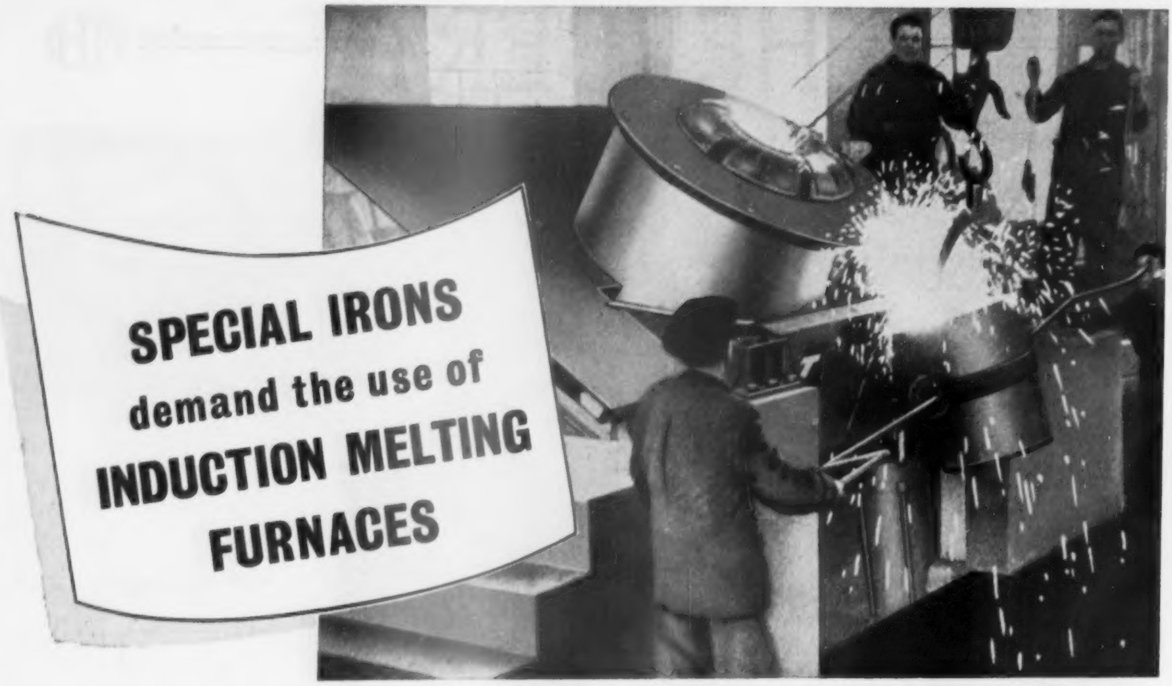

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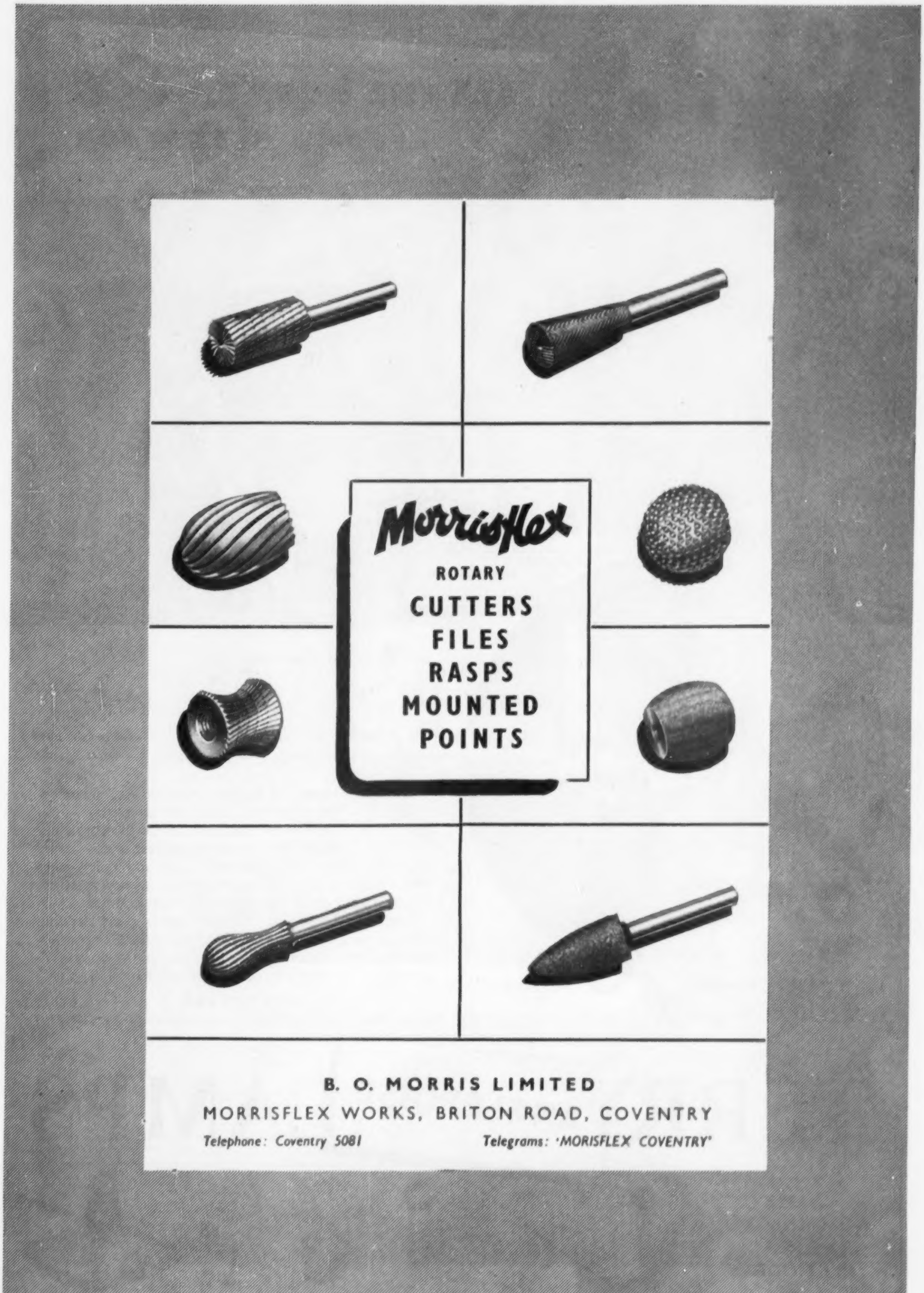
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G 124

AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

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Exports

RECENTLY there has been in some quarters a tendency to draw unwarranted inferences from the fact that the British automobile industry is not expanding sales in export markets at anything approaching the rates at which the German, French and Italian industries are. Admittedly, official statistics show that in the first seven months of this year British exports rose by less than 4½ per cent in volume over the same period for 1954, whereas in the first six months of the year French exports of cars rose by more than 20 per cent, and Italian by more than 80 per cent. The complete figures for Germany are not yet available, but it is confidently expected that they will also show more than 80 per cent increase.

However, consideration of percentage increases in isolation does not give a true picture of the situation. It would, in fact, be a matter for surprise if the French, German and Italian automobile industries had not shown greater percentage increases than the British industry has this year. As is well known, since the war the British automobile industry has been the largest exporter of motor vehicles in the world. Therefore, it is only to be expected that percentage increases in exports year by year will be significantly less than for those countries which are only now seriously seeking for outside markets.

It is the German competition that is the most serious. The industry in that country starts with a considerable geographical advantage in relation to European markets; an advantage which would become even more marked if there should be any appreciable increase in East-West trade on the Continent. In addition, the Germans have been in the fortunate position of being able to re-equip their factories almost completely in the past few years; they can call upon the resources of a large home machine tool industry; and wage levels are lower.

It is insufficiently realized, except by a small percentage of those actively engaged in the industry, how small is the proportion of total cost over which management has really effective control. Even in the larger organizations that produce a great proportion of the components they use, the percentage of total cost under direct control of the management is something in the order of 30 per cent.

The less a manufacturing organization is integrated, the lower will be the percentage of total costs that can be controlled, and for the whole industry the figure is considerably less than 30 per cent. It is therefore a matter for some surprise that prices in this country have been held for so long, particularly in view of the increased prices for coal, steel, and services such as gas and electricity.

Without any question, the main producers in the automobile industry (in this we include many suppliers of ancillary equipment) have carried out extensive reorganization of production facilities and equipment during the past six or seven years. As a result, productivity has increased at considerably more than the average rate for industry generally. But for this, which has allowed a considerable spread of overhead charges, selling prices could not have been held for so long.

There is little the industry itself can do to reduce costs. Already the most advanced production techniques are in common use; work in progress, once a common source of great expense, is now held at a figure that would have been thought dangerously low only a few years ago; and further savings have been effected by better liaison between the vehicle manufacturer and suppliers. No one in the industry would claim that perfection has been reached, but the standard is now so high that any possible improvements would have only a minor effect on total costs.

It is therefore an inescapable conclusion that if costs are to be held—a reduction to meet foreign competition would, of course, be better—relief must come from outside the industry. The greatest relief would come from Government action to reduce the incidence of taxation, both national and local, and from increased efficiency in the basic industries and supply services, so that those essentials of production, coal, steel, electricity and gas should not, at least, advance in price beyond the present levels.

Thoughts on Automation

EVERY report received from the United States of America testifies to an ever-increasing interest in, and wider application of, the principles that are now generally referred to as automation. The automobile industry in that country is certainly leading developments so far as manufacturing engineering industries are concerned. To meet the growing demand and to prepare for further advances, American machine tool organizations are extending their plants, as also are the organizations responsible for the instrumentation called for by the new techniques.

The generally accepted reason for the introduction of automation equipment into American factories is the high wage rates that are common to all manufacturing industries

in that country. Despite the existence of small pockets of unemployment in various parts of the continent, in the U.S.A., as in this country, there are now conditions of full employment. As a natural consequence, labour has been in a very strong bargaining position and has demanded and obtained very high hourly rates and, in the automobile industry, a guaranteed annual wage.

There is, therefore, an immediate economic reason for the introduction of more and more automatic devices in order to reduce the labour force and the labour costs for a specific output. But there are also other long-term reasons of considerable cogency. These may be summed up as the desire to maintain steady increase in the *per capita* standard of living of the population. In the past 15 years, the American worker's output has increased at the rate of 2·45 per cent compounded annually; with this has gone a proportionate improvement in the *per capita* standard of living. If these rates of improvement are to be maintained, greatly increased automation will be necessary.

Demographical estimates

In round figures, the present population of the U.S.A. is 165,000,000, and the number of available workers is 64,000,000. H. F. Dever, writing in the *Financial Times*, states that demographical studies show that in 20 years time the population will have risen to 220,000,000. The labour force then available will be in the range 78,000,000-86,000,000, and if the present rates of improvement in productivity and per capita standard of living are maintained, a labour force of 84,000,000 will then be required. In other words, the available labour will be barely sufficient to maintain the rate of improvement to which Americans have become accustomed, and that only if there is no reduction in the length of the working week.

From the figures quoted, it is obvious that the ratio between total population and working population is expected to remain approximately constant in the U.S.A. over the next 20 years. In this country, the probability is that so far as effective labour force is concerned, the position will worsen, in that the available labour force will represent a smaller percentage of the total population. In so far as this is true, the case for automation is stronger in this country than in the U.S.A.

This, however, is not to say that we in this country can, or should, attempt to employ automation on the American scale. Other factors must be considered. Industrialists cannot be expected to take any real practical interest in demographic estimates of the conditions that will apply 20 years hence. With affairs as they are at present, they are much more likely to think: Sufficient unto the day is the evil thereof. They are, and rightly, primarily concerned with the next two or three years, and with how far automation will improve the prospects of success in severely competitive conditions.

Dr. Nordhof, of Volkswagen, apparently thinks it will be of little avail, but he alone of large-scale producers seems to hold this opinion, which is directly contrary to American experience. If the planned hourly output and total output are high, there is not the slightest doubt that appreciable economies can be effected. The high initial outlay for the latest types of machines may act as a deterrent, but in point of fact, automatic transfer machines will give a high hourly output for a lower capital outlay than would be required for conventional individual machines to produce the same output.

Comparative costs

Some interesting figures on this point have recently been published in *Automotive Industries*. In 1914, 162 machines were supplied to an automobile manufacturer for machining four flat surfaces on cylinder heads. The output from all the machines was 108 heads per hour. At the time, the machines cost 243,000 dollars, which is equivalent to 651,240 dollars at their current value. In 1949, six transfer machines were installed to give the same output. They cost 240,000 dollars, equivalent to 271,000 dollars to-day. These machines cut direct labour costs by 50 per cent. In 1954, the six transfer machines were replaced by a single machine costing 230,000 dollars. This machine reduced the direct labour costs to one-tenth of the 1914 figures.

CONTENTS

THE FIAT 600

Part I: The 633 cm³ Engine—A Light Weight, Compact Unit of Sturdy Design

THE Fiat 600 is an outstanding post-war example of the successful design to suit a specific market. The vehicle has been designed primarily to satisfy customers in Italy, where petrol costs the equivalent of about 7s. 3d. per gallon and, as a consequence, economy of fuel consumption is of major importance. This has involved the employment of as small an engine as possible. To do this without any appreciable sacrifice in acceleration or top speed, it has also been necessary to make the vehicle as light as possible.

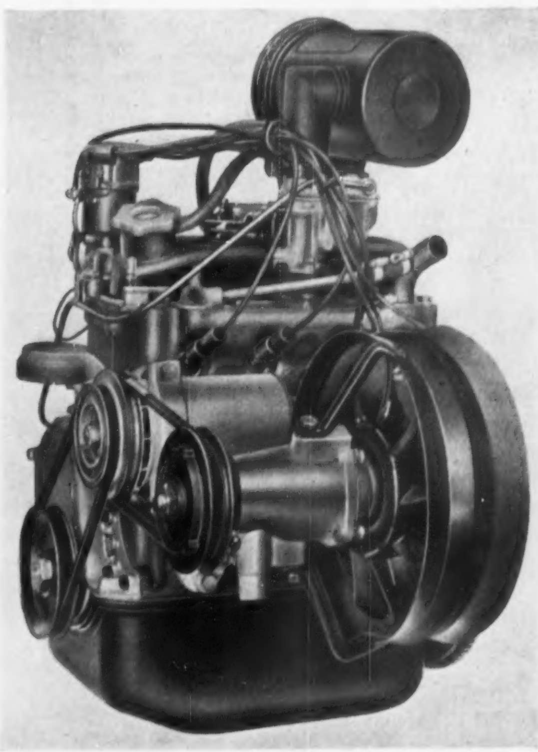
SPECIFICATION

ENGINE: Four cylinders. Bore and stroke 60 mm and 56 mm. Swept volume 633 cm³. Maximum b.h.p. 21·2 at 4,600 r.p.m., without silencer, fan and water pump. Maximum b.m.e.p. 113·7 lb/in² at 2,800 r.p.m. Maximum torque 347·2 in-lb, without silencer, fan and water pump. Compression ratio 7:1. Three-bearing crankshaft, not balanced dynamically. Overhead valves, push-rod and rocker operated. Weber 22 DRA carburettor with 15·5 mm diameter choke. Diaphragm type, mechanically operated, fuel lift pump.

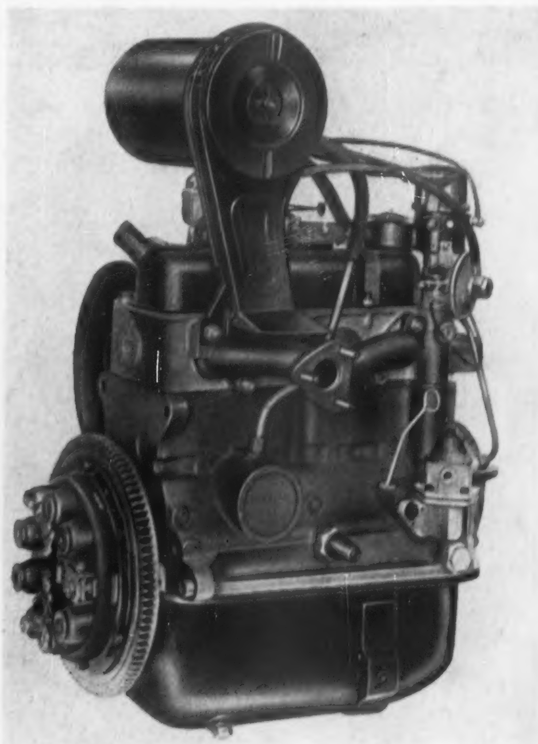
After careful investigation, the manufacturers decided that, for a given space for passengers, minimum overall dimensions cannot be obtained with the conventional layout, that is, with the engine at the front and final drive at the rear. Accordingly, the rear engine layout was adopted. With this arrangement, the overall length of the vehicle can be kept to a minimum because the toe-board can be further forward than if the engine is at the front. Moreover, the weight and cost of a propeller shaft can be saved; even though a tunnel is still used as a structural member in the Fiat 600, it does not occupy as much space in the body as would a tunnel deep enough to accommodate a propeller shaft.

Further saving in weight and space is obtained by grouping the final drive and gearbox together in one casing. A reduction in overall length has also been obtained by positioning the radiator beside the engine. One of the most noteworthy features of this design is the way in which independent suspension all round has been obtained with a layout having the simplicity that is essential if manufacturing costs are to be kept down to the low level necessary for such a small car. In short, careful analysis of the requirements, and intelligent design to meet them, has resulted in a vehicle that has aroused great interest and admiration.

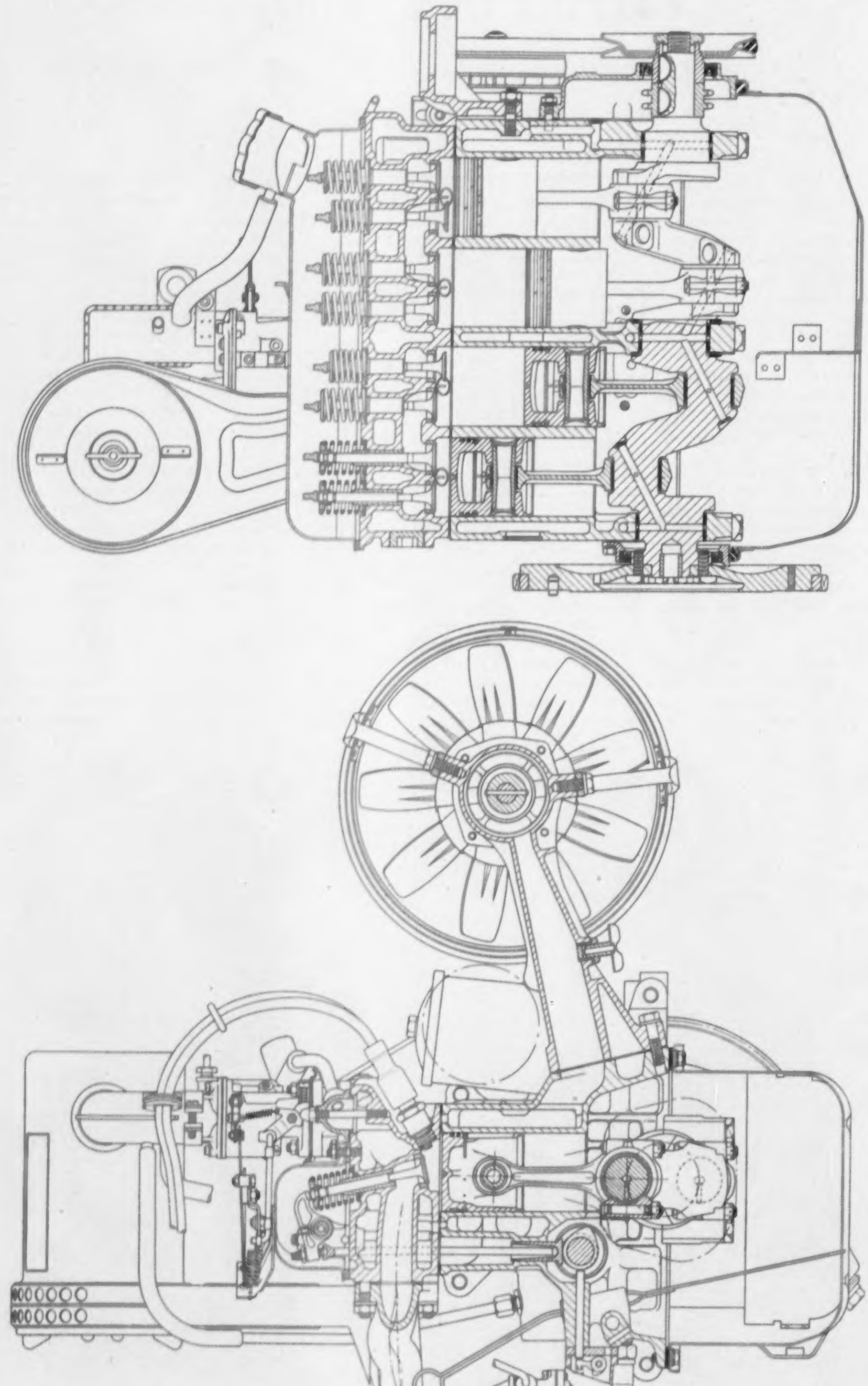
That such a well engineered design should come from the Fiat organization is not surprising, for their manufacturing experience is extensive. This organization produces private cars and commercial motor vehicles, industrial and farm tractors, railway and tramway rolling stock, diesel marine and stationary engines of all sizes, as well as aviation engines and aircraft, machine tools, and refrigerators. They also make a wide range of lubricating oils and greases for all purposes. The organization is of the vertical type: many of the key basic materials are mined by Fiat and the primary processing of these materials, as well as the manufacture of components and



A two-stage vee-belt drive is taken from the rear of the crankshaft to the dynamo and thence to the water pump and fan unit



A control is fitted to allow the air to be drawn into the air cleaner either directly, or from over the exhaust manifold

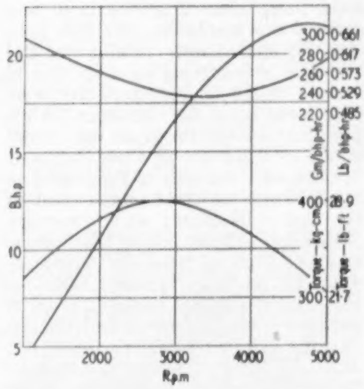


ARRANGEMENT OF THE FIAT 600 ENGINE
Bore 60 mm. Stroke 56 mm. Swept volume 633 cm³

final assembly, is also done by them. In fact, as far as cars are concerned, all except the trim, glass and brake linings are made by Fiat or their subsidiaries. The factory at which the Fiat 600 is made is at Mirafiori, Turin, where 22,000 people are employed in two shifts.

Engine, general layout and performance

Since the engine is installed at the rear, the transmission drive is taken from the front end of the crankshaft and the timing drive from the rear end. The layout of the unit is fairly conventional, except that the radiator, and therefore the fan and water pump, are on the right-hand side, as viewed from the rear. Thus, instead of the usual triangulated drive, a two-stage vee-belt drive is taken from the tail end of the crankshaft. Its first stage is to the dynamo, mounted on the side of the cylinder block, and the second is from the dynamo to a pulley on the rear end of the spindle of the water pump and fan assembly. The bracket that carries the water pump and fan unit is hollow and forms part of the ducting of the coolant system, since it directs water from the pump outlet into the base of the cylinder block. An oil filter is also mounted on the right-hand side of the engine; it is situated between the crankcase and the ducting that takes the air from the fan to the radiator, and is more or less in line with the web that carries the intermediate main journal bearing.



Engine performance, without the fan and silencer. The power and torque curves are subject to a tolerance of −5%, and the specific fuel consumption curve to +5%

The exhaust manifold is on the left-hand side of the engine, while the inlet manifold is on the right and is cast integrally with the aluminium cylinder head. Among other components on the left-hand side of the engine are the petrol pump, distributor and contact breaker unit, and the oil pressure gauge connection, which is screwed into the oil gallery. The oil pump is bolted up to the lower face of the rear wall of the crankcase casting, and is driven in the conventional way by the shaft that serves the contact breaker and distributor unit.

A stroke:bore ratio of 0·933:1 has been adopted and the connecting rod length:stroke ratio is 1·857:1. At the engine speed at which maximum b.h.p. is developed, the mean piston speed is 1,690 ft/min. The ratio of maximum torque to the torque developed at maximum b.h.p. is 1·194:1, and the ratio of the speeds at which maximum torque and maximum b.h.p. are developed is 0·608:1. A maximum b.m.e.p. of 113·7 lb/in^2 is obtained at 2,800 r.p.m. At 3,200 r.p.m. the i.m.e.p. is 136·5 lb/in^2. The minimum brake specific fuel consumption is 0·578 pt/b.h.p-hr.

The dry weight of the engine is 154·3 lb, so the b.h.p. developed per lb is 0·137. This is a particularly good output for this type of engine and undoubtedly reflects the judicious use of aluminium alloys and the general compactness of the unit. The other specific output figures include 1·21 b.h.p/in^2 piston area and 33·6 b.h.p/litre. These figures are normal in view of the fact that the piston area is large, because of the low bore:stroke ratio, and also that since the head is of aluminium alloy, a generous thickness of metal has to be allowed round the valve seats. In general, the engine is moderately rated and from this viewpoint there is every reason to suppose that the unit will give long and trouble free service. The overall dimensions of the engine are: height, less air filter, $21\frac{11}{32}$ in; width, less air filter, $25\frac{5}{16}$ in; length, less flywheel, $16\frac{21}{32}$ in. Rubber mountings are employed. A single, high-point type of layout has been


Above the timing drive, a single high-point mounting is bolted to the rear end of the cylinder block

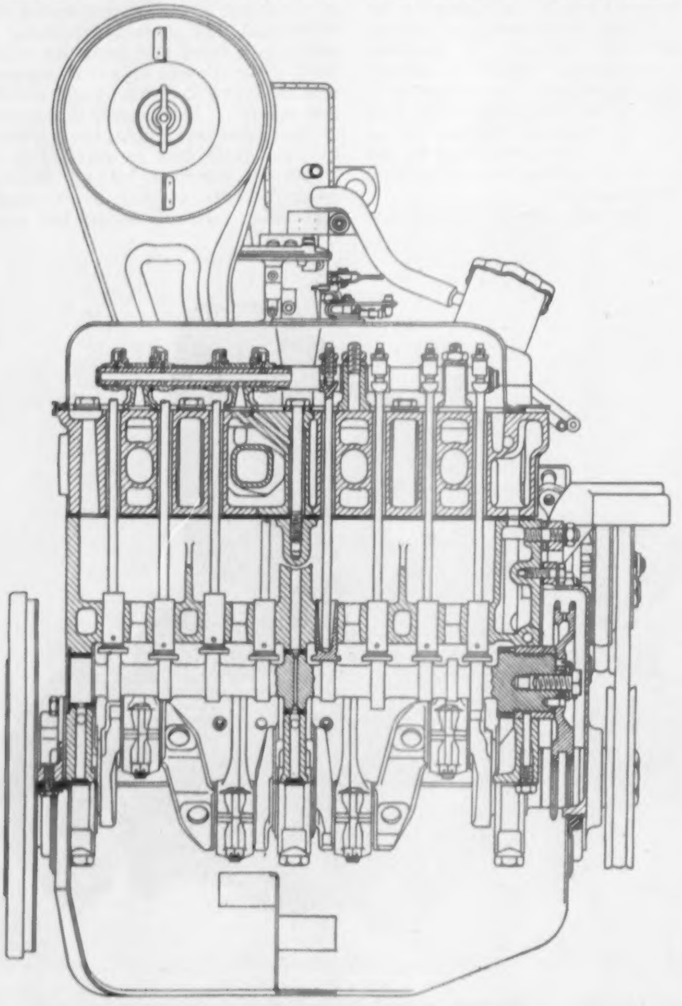

The seals between the ends of the crankcase and the sump are outboard of the main journal caps

adopted for the rear mounting, which is of conical form with its axis vertical. Two sandwich type units, one on each side of the gearbox, are employed at the front. The engine is installed at an angle of 3 deg 30 min from the horizontal.

Cylinder block and crankcase

The integral cylinder block and crankcase is of grey cast iron. In the interest of compactness, the front pair of cylinders is siamesed, and so also is the rear pair. This arrangement has not led to distortion problems; presumably this is because of the small size of the cylinders, the moderate power output, and the good thermal conductivity of the aluminium alloy cylinder head. At the centre of the block, the space between the adjacent cylinders is $\frac{3}{32}$ in. A cylinder wall thickness of $\frac{5}{32}$ in has been adopted, except at the junctions between the siamesed pairs of cylinders, where it is $\frac{11}{32}$ in. The top decking of the block is $\frac{11}{32}$ in thick, and the side walls are about $\frac{5}{32}$ in thick.

Unconventional sealing arrangements have been adopted at the front and rear of the crankcase. At the front, that is, immediately behind the flywheel, a housing ring for a lip type oil seal is bolted to the crankcase wall. The inner lip of the seal bears on the periphery of the flange on the crankshaft, to which the flywheel is bolted. In this way, the area to be ground to a fine finish to avoid undue wear of the seal is restricted to a minimum. Shoulders on each side of the housing ring form an extension of the sump joint-face on the crankcase, and the joint is continued round the outer periphery of the lower half of the housing ring by a semi-circular, rubber seal. This seal is housed in a channel formed by welding a Z-section to the edge of the front face of the sump, where a semi-circular cut-out clears the seal housing. The $\frac{3}{32}$ in thick, sump face joint washer extends over the ends of the semi-circular rubber seal. A similar arrangement is employed at the rear, but the housing for the lip type seal round the crankshaft is formed in the timing cover. Thus, both the front and rear main journals are totally enclosed in the crankcase and sump.



Location of the camshaft is effected at its rear bearing

As compared with the more conventional method of sealing round the bearing caps, this method of sealing has the advantage that it avoids the necessity for the provision of special return arrangements for oil that issues from the outer ends of the bearings. These oil return arrangements are generally complicated because the lubricant has to pass the rapidly rotating crank webs as it drains to the sump. The alternative of employing a deep skirted crankcase and sealing on a bridge piece under each end bearing, is only attractive for very highly rated engines that require exceptionally stiff crankcases. This is because the machining operations on deep skirted designs are more costly than on crankcases of the type employed on the Fiat 600 and most other quantity produced engines for small family saloon cars.

The walls of the crankcase extend down to a level approximately $\frac{6}{32}$ in below that of the axes of the main journals, and the lower faces of the central web and end walls are recessed to the same depth, that is, $\frac{6}{32}$ in, to form the seatings for the main journal caps. These caps are located laterally between the shoulders at the ends of the seating faces. They are retained by 0·3937 in diameter set-bolts, of a material similar to En 19A or En 25, and are locked by tab plates. A torque spanner set to 538 in-lb is employed to tighten the bolts.

Crankshaft

A three bearing, forged steel crankshaft of a material similar to En 8K is employed. The main journals are 1·996 to 2·00039 in diameter. Between the outer faces of the end webs, the length of the crankshaft is $10\frac{13}{32}$ in. The webs are $\frac{1}{2}$ in thick by $2\frac{3}{4}$ in wide, and the inclined arms of the cranks are $1\frac{11}{32}$ in by $2\frac{1}{16}$ in cross section. All the crankpins are 1·3775 - 1·3783 in diameter. Axial location is effected at the intermediate bearing, where Vandervell D2 thrust washers are carried in a counterbore in each end of the bearing housing and cap. The axial float allowed is 0·00236-0·010236 in.

Vandervell D2 Bimetal bearings are employed at the front and rear. Their effective lengths are 0·80197-0·81197 in. At the centre, an unusual arrangement has been adopted. The half shell in the crankcase is a D2 Bimetal component, while the other half, in the cap, is a Trione, copper-lead lined, steel backed bearing shell. This arrangement has been adopted because the crankshaft is unbalanced and, therefore, the loads on the centre bearing are relatively heavy. To make use of existing standard shells, the Vandervell half is 0·8869-0·89697 in long, while the Trione half is 0·8968-0·8868 in long. The diametral clearance in all the main journal bearings is 0·00236-0·00059 in.

It is of interest to compare this shaft with that of the Austin A30 described

in the October 1952 issue of the *Automobile Engineer*. The reason why an unbalanced shaft has proved satisfactory in the Fiat and not in the Austin engine undoubtedly is because of the relatively short stroke and large diameter bearings of the Fiat engine. These two features have led to a large overlap between the main and big end bearings, which must have contributed much to the stiffness of the crankshaft. The relevant dimensions of the A30 crankshaft are: stroke 76 mm, main journal diameter $1\frac{3}{4}$ in, crankpin diameter $1\frac{15}{32}$ in, and length between the outer faces of the end crank webs $10\frac{27}{32}$ in. It is also of interest to note that the bearing areas in the Fiat engine are smaller than those in the Austin, in which the bearings are all $1\frac{3}{16}$ in long.

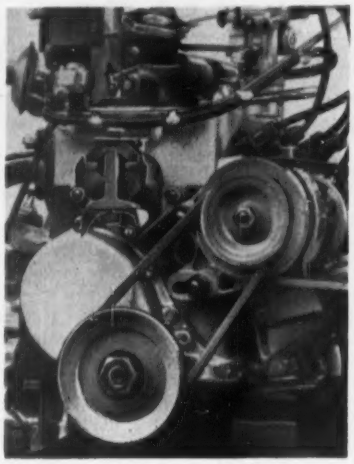
A grey cast iron flywheel is employed. Its overall dimensions are: diameter $9\frac{1}{4}$ in, thickness $\frac{11}{16}$ in. The assembly, complete with the starter ring gear, weighs 9·325 lb and has a mass moment of inertia of 35 ft-lb-sec^2. Six 0·315 in diameter set screws secure the flywheel to a flange round the crankshaft, on the end of which it is spigoted. The induction hardened carbon steel ring gear, which has 97 teeth, is pressed on. An $\frac{11}{16}$ in diameter counterbore is drilled in the end of the crankshaft to house the $\frac{7}{16}$ in inside diameter phosphor bronze bush that carries the clutch shaft. A trap is formed in the rear face of the flywheel to prevent lubricant flung out from this bush from reaching the friction facings of the clutch. The trapped lubricant is carried away through holes to the rear face of the flywheel.

Connecting rod and piston assembly

H-section connecting rods of a material equivalent to En 5K are employed. Their centre-to-centre length is $4\frac{25}{32}$ in, and their minimum cross sectional dimensions are about $\frac{7}{16}$ in deep by $\frac{3}{8}$ in wide over the flanges by $\frac{1}{8}$ in thick. Each rod is offset $2\frac{1}{2}$ mm from the centre of the big end bearing in a direction towards the adjacent main journal. This is necessary because of the close spacing between No's 1 and 2, and between No's 3 and 4 cylinders.

Each rod weighs $\frac{3}{4}$ lb. The big end bearing caps are split at an angle of 90 deg; nevertheless, because of the large bore:stroke ratio, the diameter of the cylinders is large enough for the rods to be withdrawn through them. Lateral location of the caps is effected by the $\frac{5}{16}$ in diameter bolts, which are of a material equivalent to En 19A or En 25. Self locking nuts are employed. They are tightened with a torque spanner set to 199·6 in-lb.

The big end bearings are Trione - Botto, copper - lead lined, steel, shell type components. They are 0·82795-0·83779 in long and their diametral clearance is 0·002244-0·0004724 in. An interference fit in the housing of 0·002086-0·0009488 in has been adopted. A wrapped bronze bush is fitted in the small end. It is 0·68897-0·70078 in long and its diametral clearance is 0·0005118-0·0000787 in. The gudgeon pin is case hardened to 178 to 255 B.H.N. The inside diameter of the pin is tapered from 0·433 to 0·5118 in. To allow for the thermal expansion of the aluminium alloy of the piston, the pin is an interference fit of 0·0000787-0·000472 in in the cross-holes. Axial location of the pins is effected in the conventional manner by wire circlips in the ends of the cross-holes.



The rear mounting on this engine has been sectioned to show the arrangement of its components

Fiat manufacture the aluminium alloy pistons, which are of a specification similar to B.S.1490 LM-2-M. They are of the T-slotted type with flat crowns, and weigh 0·39 lb. Two Nova compression rings are fitted in each piston. Their principal dimensions are: gap 0·00787-0·013779 in, face width 0·09756-0·0980 in, radial thickness 0·09527-0·10157 in. The depth of the compression ring grooves in the piston is 0·10354-0·10748 in, and the side clearance between the rings and the grooves is 0·00177-0·00283 in. A U-Flex oil control ring is employed. Its face width and radial thickness are 0·15397-0·155 in and 0·12799-0·14 in respectively. The depth of its groove in the piston is 0·164-0·168 in and the side clearance is 0·000984-0·002047 in.

Timing gear, camshaft and valve gear

A B.S.1490 LM-4-M timing cover is bolted to the rear face of the crankcase and houses the lip type oil seal, which bears on the periphery of the boss of the vee-belt drive pulley. The pulley and the drive sprocket are keyed-on and retained by a nut and tab washer on an $\frac{11}{16}$ in diameter thread on the $1\frac{1}{4}$ in diameter extension of the crankshaft. The drive is transmitted by a $\frac{3}{8}$ in pitch two-strand chain from the En 8K sprocket to the half-speed wheel.

A single bolt, locked by a tab washer, is passed axially through the half-speed wheel to retain it on a spigot on the front end of the camshaft. The arrangement of the camshaft bearings is unusual for this type of engine. Sintered hard bronze bushes are employed for the front and intermediate bearings, and a copper-tin alloy bush of relatively large cross section is fitted at the rear. This rear bearing is located by a dowel ended set-screw assembled into a hole drilled vertically from the bottom of the crankcase, and registers in a radial hole in the bush. As a result of the adoption of this arrangement, the front face of the cylinder block is flat; consequently, the machining operations on it are as simple as possible. In addition, the assembly has been simplified by the elimination of the thrust plate bolted to the front face of the block, which is the more conventional arrangement.

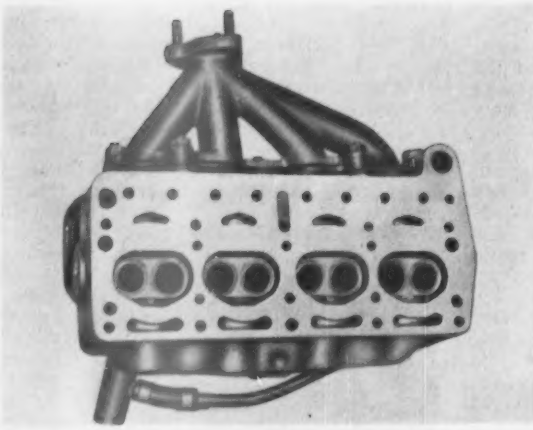
The dimensions of the rear bearing bush are: inside diameter 1·497 to 1·4986 in, outside diameter 1·8898 to 1·8888 in, length 1·0827 to 1·0814 in. To facilitate assembly of the camshaft from the rear, the intermediate bearing is of

VALVE DATA

	Inlet	Exhaust
Material	En 100 C	X B
Head diameter	$\frac{11}{16}$ in	$\frac{7}{8}$ in
Throat diameter	$\frac{21}{32}$ in	$\frac{3}{4}$ in
Stem diameter	$\frac{5}{16}$ in	
Diametral clearance in guide	0·000866—0·002165 in	
Seat angle	45 deg 30 min	
Seat material	grey cast iron	
Spring material	En 42	
Spring rate	68·62 lb/in	
Spring length, free	$2\frac{1}{32}$ in	
Spring length, installed	$1\frac{17}{64}$ in	
Spring surge frequency	22,000 c.p.m.	
Number of coils	7·5	
Coil diameter	0·7953 in	
Wire gauge	0·1181 in	
Valve lift	0·295 in	0·2768 in
Rocker ratio	1·5	
Valve crash speed	6,500 r.p.m.	
Valve guide material	cast iron	
Valve guide length	$1\frac{13}{16}$ in.	
Valve guide inside and outside diameters	$\frac{5}{16}$ in $\frac{17}{32}$ in	
Tappet clearance, cold; for running	0·0039 in	
for timing	0·0083 in	
Valve opens	10 deg B.T.D.C.	35 deg B.B.D.C.
Valve closes	35 deg A.B.D.C.	2 deg A.T.D.C.
Ignition timing:		
Initial advance	10 deg	
Automatic advance	30 deg	
Vacuum advance	11 deg	
Total max. advance	40 deg	

a smaller diameter and the front bearing is smaller still. The front and intermediate bushes are of sintered, hard bronze and are 0·59055 in long. At the intermediate station, the bush is 1·4996-1·5008 in inside diameter by 1·6535 - 1·6547 in outside diameter, and that at the front is 1·2236 - 1·2248 in inside diameter by 1·4169-1·4181 in outside diameter. The diametral clearance in the front and intermediate bearings is from zero to 0·002953 in, and that at the rear is 0·000984-0·003504 in.

Between the bearings and cams, the shaft, which is of cast iron and is produced by the shell moulding process, is 0·82677 in diameter. The cams are profiled to give a maximum positive acceleration of the tappet of 7,874 ft/sec^2 and a maximum negative acceleration of 2,624 ft/sec^2 at 4,600 engine r.p.m. A lift of 0·19929 in has been adopted for the inlet and 0·187165 in for the exhaust cams. The nominal period of the inlet cam is 76 deg and of the exhaust 74 deg. A gear to drive the oil pump and distributor is incorporated immediately in front of the rear bearing. It has 12 teeth set at a helix angle of 66 deg and has a pressure angle of 20 deg. The eccentric that actuates the fuel lift pump is between the rearmost pair of cams.


The ducts through which coolant passes into the cylinder head are immediately under the sparking plug bosses and exhaust valve ports

Cast iron tappets of the piston type are employed. The diameter of the holes in which they are housed in the cylinder block is 0·55157-0·55228 in and the diametral clearance is 0·00181-0·000394 in. Carbon steel push rods, the ends of which are induction hardened to 178-248 B.H.N., seat in the tappets. Their effective length is 7$\frac{3}{32}$ in and their diameter $\frac{1}{4}$ in. A spherical end is formed at the bottom of each rod and a cup is up-ended at the top to receive the spherical end of the tappet adjusting screw.

The end pads of the carbon steel rockers are induction hardened to 178-248 B.H.N., as also are their bores. Instead of the usual screw-driver slot in the ends of the tappet adjusting screws, two flats are formed diametrically opposite one another on each to receive a spanner. The setting is locked in the conventional manner by lock-nuts.

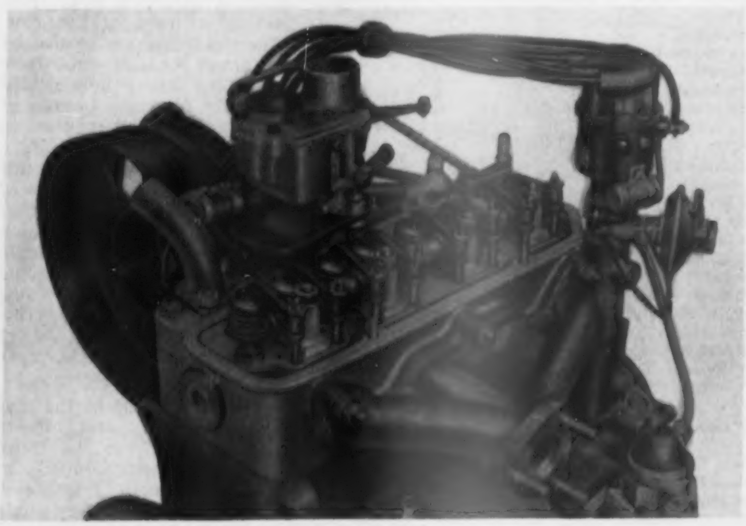
A carbon steel rocker shaft, case hardened to 178-255 B.H.N., is carried on four B.S.1490 LM-4-M aluminium alloy rocker pedestals. It is about $\frac{11}{16}$ in outside diameter by $\frac{11}{32}$ in inside diameter. The effective bearing length in each rocker is 0·59-0·582 in, and the diametral clearance is 0·0006299-0·002165 in.

The rockers are assembled one in front of, and another behind, each pedestal. An unconventional feature of the arrangement is that distance tubes are fitted instead of coil type compression springs between the rockers, and at each end, the assembly is retained by a circlip and plain washer. Presumably, this arrangement has been adopted because the tubes are less expensive than springs, and there is no need to locate the rockers with extreme accuracy since their end pads are wide enough to allow for a certain amount of axial float. In any case, the pads are offset a distance of about 1 mm from the centres of the ends of the valve stems. This arrangement has been adopted, of course, to induce a certain amount of valve rotation.

In each cylinder, the valves are spaced with their axes about 1$\frac{3}{16}$ in apart. So far as length, inside and outside diameters are concerned, the valve guides are all identical. Their upper ends are tapered to form a lead-in to facilitate assembly into their housings. Positive axial location is effected by circlips in grooves round the guides. These circlips seat in counterbores in the upper ends of the guide housings and are retained by flat washers on which the springs seat. This arrangement is better than an integral collar round the guide to provide axial location, because it enables the components to be finished to size by centreless grinding.

The lower ends of the guides are flush with the holes in which they are housed. In all the ports this gives a minimum of obstruction to the gas flow. However, in the exhaust ports, the greatest benefit is obtained by virtue of the fact that the end of the guide is well cooled and is shielded from the hot gases. Therefore, carbon does not tend to form in the bore. In addition, the portion of the valve stem that moves into the hot gas stream when the valve is opened, is undercut by 0·022 in, as measured on the diameter, so that it does not contact the bore of the guide when the valve closes. This helps to avoid the build-up of carbon due to the burning of the oil film on the portion of the stem that is alternately in the guide and in the exhaust gas stream.

Circlips are fitted in grooves round the valve stems above the guides, to prevent the valves falling into the cylinder in the event of a spring breakage. An unusual feature of the design is that the clearances for both the inlet and the exhaust valves are identical. The springs are retained in the conventional manner by washers and split tapered collets. It has not been found necessary to fit rubber rings in the valve collets to prevent oil from running down the stems. Further details of the valve assembly are


Distance tubes, instead of the more conventional coil springs, are fitted between the rockers

given in the accompanying table.

Cylinder head

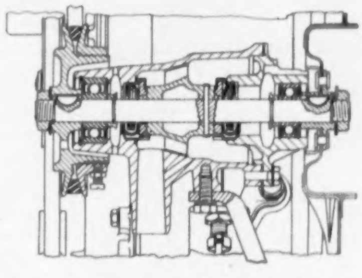
An aluminium alloy, equivalent to B.S.1490 LM-4-M, is employed for the cylinder head. As has already been mentioned, the casting incorporates the induction manifold. The overall dimensions of the head are: depth $2\frac{11}{16}$ in, width $6\frac{1}{4}$ in, length $13\frac{11}{16}$. It is held down by ten $\frac{7}{16}$ in diameter studs of a material equivalent to En 19A or En 25. The nuts are tightened with a torque spanner set to 260·4 in-lb. A Klingerit asbestos washer, reinforced with steel tape at the edges, is employed as a gasket, and a cork washer is interposed between the pressed steel rocker cover and the head.

By placing the inlet manifold on one side of the head and the exhaust on the other, it has been possible to improve the efficiency of, and also to simplify, the breathing arrangements. A cast iron exhaust manifold is employed and, as can be seen from the accompanying illustrations, the branch pipes are relatively long and straight. Their inside diameter is $\frac{11}{16}$ in. They are all joined at their inner ends to a common flange secured to the cylinder head by five studs. The outer ends of the branches are joined to a three-stud flange that receives the down pipe. As can be seen from the illustration of the engine installation, this pipe sweeps down in a gentle curve to a large silencer to the left of the engine sump.

Carburettor and induction system

A Weber, 22 D.R.A. downdraught carburettor is employed. Its choke diameter is 15·5 mm. The jet sizes are as follows: main jet 0·87 mm, starting jet 1·05 mm, idling jet 0·45 mm, slow running air bleed, 2·30 mm, needle seating 1·75 mm. A diaphragm type mechanical pump draws the fuel from the 5·94 gallon tank, which is installed under the bonnet lid at the front. This arrangement should be a good one from the point of view of vapour lock, since the tank is relatively high and is in a cool place. In more conventional layouts, where the tank is near the tail end of the exhaust pipe, it has been known for the exhaust gases to swirl round the exterior of the tank and to raise the temperature of the petrol sufficiently to cause vapour lock.

An unconventional form of filter, with a Fispa, sprayed-cloth element, is employed. It is of cylindrical shape, installed with its axis horizontal, and the pipe to the carburettor inlet is taken from the side. This reduces the overall height of the unit by comparison with the more usual arrangement, with the pipe to the carburettor inlet taken straight down from the base of the filter. Another unusual feature is that provision is made either to take cool air directly from the filter or to draw in warm air from over the exhaust manifold. A simple hand-control is incorporated to effect the change. This control comprises a sleeve-valve that can be rotated inside one end of the cylindrical unit. Normally, a port in the sleeve valve coincides with a number of holes round the upper part of the periphery, adjacent to the end of the housing, so that air can enter through these holes. However, if the valve is turned through 180 deg, it blanks off these holes and the port is then in line with the intake from the duct through which the air from over the exhaust manifold is drawn. This arrangement is necessary because in rear-engined installations, particularly where the radiator is to one side and, therefore, warm air does not flow over the induction system,



Section through the coolant pump, showing alternative water seals

carburettor icing trouble may be experienced.

The carburettor is mounted on a short down pipe, flanged at each end and bolted on top of the head. It is of interest to note that although the unit is remote from the exhaust manifold, a metal shield is interposed between the float chamber and the engine. This is because in Italy there are many mountainous areas, and during the summer, the climate is relatively warm. In these circumstances, vapour lock is likely to be experienced unless precautions are taken to prevent it. The down pipe in the head joins a longitudinal induction tract, which is water jacketed. Two ports, each serving two inlet valves, are taken from the track. Since the down pipe is midway between the two ports, both should get the same quality of mixture.

Water pump and coolant system

A nylon-reinforced rubber fan-belt, made by Pirelli, is employed. It has a vee-angle of 36 deg and is $\frac{1}{2}$ in wide by $\frac{11}{32}$ in depth. The pulley is driven at $1\frac{1}{2}$ times engine speed. In the engines now being produced, the pulley comprises an aluminium alloy hub, on to which are bolted two pressed steel flanges with spacer rings between them. These spacer rings form the base of the channel between the flanges, and one or more can be removed or added to adjust the belt tension.

As has already been stated, the fan and pump assembly is mounted on a bracket that also forms the water outlet to the cylinder block. From the illustration it can be seen that the bore of the passage to the block is shaped in such a way as to change the velocity head into a pressure head. A drain plug is fitted in the base of this passage. The plug is unusual in that, to open the drain, it is screwed radially into the passage instead of out. This is because the water drains into a diametral hole near its inner end and thence out through an axial hole that is drilled from its outer end to break into the diametral one. The plug has a brazed-on wing type head, so that it can be turned by hand without a spanner. At the lower end of the bracket is a flange for the three set-bolts that secure it to the cylinder block and crankcase unit, and the pump body is formed integrally with its other end.

A cast iron rotor, $2\frac{3}{32}$ in diameter, is assembled on to the centre of the $\frac{11}{16}$ in diameter spindle, which is of En 15A

To save space, the power unit is installed with the radiator on its right-hand side

or En 111. The rotor is secured by a rivet passed diametrically through its boss and the spindle. It is of the axial- rather than the radial-flow type and doubtless is a highly efficient unit. At each end of the rotor there is a Morgan SD 45 seal. Each seal is of the spring-loaded rubber type and a carbon thrust ring is interposed between it and the end of the rotor. The front seal is housed in the pump casing, but the rear one is carried in a cast aluminium alloy housing spigoted into the end of the pump casing. This housing also carries the rear bearing. A drainage space, in which a thrower ring operates, is incorporated between the seal and the bearing. This ring is assembled over the tail end of the spindle, followed by the sealed ball bearing, and then the keyed-on fan hub. The whole assembly is pulled against a circlip in a groove round the spindle by a nut on the front end. The outer race of the bearing is not positively located axially in its housing.

Another sealed ball bearing is employed at the rear. Its centre is about $5\frac{3}{64}$ in in front of that of the front bearing. The outer race is positively located by a snap ring in its housing in the nose piece of the pump casing. As at the front, a drainage space is incorporated between the water seal and the bearing. The assembly on the rear end of the spindle also is similar to that at the front: it comprises a thrower ring, the sealed bearing and the vee-belt pulley, which is keyed on and retained by a nut on the end of the spindle.

The fan is $9\frac{1}{16}$ in diameter and has nine pressed steel blades. Its shroud ring is mounted on three stays bolted to the pump casing. This ring is of venturi form to smooth out the air flow, and is a clearance fit in the duct to the radiator to allow for movement of the engine relative to the radiator assembly, which is bolted to the bulkhead. The outlet duct from the radiator incorporates a thermostatically controlled shutter to regulate the engine temperature. This control is adjusted so that the shutter opens gradually after the water reaches about 75 deg C and is wide open when the water temperature exceeds 100 deg C, when a red light is illuminated on the instrument panel.

The coolant system is pressurized to the unusually high level of 22·7 to 25·6 lb/in². In this way, and by efficient ducting, the overall dimensions of the radiator have been reduced sufficiently to enable it to be installed alongside the engine. Since, with a small radiator, the water temperature is relatively high, it has also been practicable to use the radiator as the heater element for warming the air supply to the interior of the car. The area of the radiator is 0·802 ft², and the block thickness is $2\frac{11}{16}$ in.

Water passes from the base tank to an inlet on the bottom of the pump casing. The bore of this inlet pipe is tapered to change the pressure head to a velocity head and so to reduce shock at the entry to the rotor. In addition, the passages between the vanes of the rotor are designed to give efficient operation and freedom from cavitation. The employment of the axial flow rotor keeps the diameter of the unit and, therefore, the dead area round the hub of the fan, to a minimum. A relatively small diameter hole connects the inlet with the delivery side of the pump to relieve the suction in the inlet when the engine is running at high speeds, thus avoiding cavitation.

Coolant delivered by the pump passes through the passage in the support bracket to the base of the jacket round the cylinders. It then passes up into the head through ducts under the sparking plug bosses and the exhaust ports. In this way the hottest areas are directly swept by cooling water. To ensure uniform distribution of coolant flow throughout the block and head, there are two outlets. Both are on top of the head, but one is near the front and the other near the rear. The rearmost one is more or less directly above the inlet, and so is restricted in size. It is in the form of a domed casting spigoted into the top face of the head and secured by a domed nut on a single, central stud. From this dome a $\frac{1}{2}$ in outside diameter pipe takes the water to the main outlet, which is a flanged pipe, $1\frac{5}{32}$ in diameter, bolted near the rear end of the head.

Oil pump and lubrication system

The gear type pump, driven at half engine speed, is bolted to the lower face of the rear wall of the crankcase. Oil is drawn through a gauze strainer of 8·6 in² area in the pressed steel sump, which has a capacity of 4·92 pints, to the pump and thence to a vertical duct in the end wall of the crankcase. From this duct a transverse passage communicates with the pressure relief valve, which is screwed into the left-hand side of the crankcase. This relief valve is set to blow off at $35\frac{1}{2}$ to $42\frac{1}{2}$ lb/in² under normal running conditions.

At the upper end of the vertical duct from the pump, the lubricant passes into the $\frac{9}{32}$ in diameter gallery, which also is on the left-hand side of the crankcase. From the gallery a passage is drilled transversely through the intermediate web of the crankcase. At the right-hand end of this passage is the by-pass type oil filter, and at the left-hand end is the oil pressure gauge.

From the centre of this transverse passage, oil passes downwards through a vertical duct and a hole in the intermediate main journal bearing shell, into a groove round the bearing surface. The journal is drilled $\frac{11}{16}$ in diameter diametrically so that lubricant can pass from the groove into the centre of the journal. Two more holes, one on each side, drilled diagonally through the adjacent crank pins, inclined arms and webs, carry the oil from the centre of the main journal into the adjacent crankpins. The outer ends of these two holes, where they break out of the inclined crank arms, are sealed with peened-in plugs. A radial hole, $\frac{1}{8}$ in diameter, in each crankpin feeds the oil to the big end bearing. These radial holes are drilled from lightly loaded areas on the crankpins.

In the front wall of the crankcase, a transverse hole is drilled from the gallery to a point above the main journal bearing, where it is joined by a vertical duct to the groove round the bearing surface. The adjacent big end bearing is lubricated by oil passing through a diametral hole in the journal, into a duct drilled through the crankpin to break into this hole, and thence out through a radial hole to the big end bearing. The rear journal and big end bearings are lubricated in a similar manner, but the layout of the ducts is different. From the rear end of the gallery, a hole is drilled vertically to join a transverse passage above the rear bearing of the camshaft. A duct from the right-hand end of the transverse hole passes the oil down to the rear journal bearing.

From the centre of the transverse passage above the camshaft, a vertical duct passes oil vertically downwards to the rear bearing of the camshaft. A spreader groove is machined longitudinally in the camshaft journal. This groove also serves to connect, once every revolution, the oil feed to a small jet drilled through the rear end of the bush to squirt oil into an annular cavity round the boss of the half-speed wheel. From this cavity the lubricant is flung outwards through radial holes to the chain.

The other two camshaft bearings are lubricated by $\frac{3}{32}$ in diameter holes drilled directly from the gallery. Two flats are machined diametrically opposite to one another in the camshaft intermediate journal bearing, and a hole is drilled through the centre of the journal to connect the two. This provides for an intermittent supply of oil to a vertical duct and thence to one of the intermediate rocker pedestals. The lubricant then passes up through the pedestal into the hollow rocker shaft and thence through radial holes to the rocker bearings. Ducts are drilled from above vertically into these bearings to pass lubricant out of the top faces of the rockers, whence it runs down into the cup ends of the push rods. Finally, it drains to the tappets and then back to the sump.

NOVEMBER ISSUE

THE description of the Fiat 600 will be completed in the November issue of this journal. It will deal in detail with the mechanical components of the chassis. An authoritative review of the exhibits at the Frankfurt Show will also appear in this issue.

CHAIN BROACHING

A New Technique Developed by the American Colonial Broach and Machine Company

A MACHINE that employs a completely new technique for surface broaching has recently been developed by the American Colonial Broach and Machine Company. This machine, which is designated the "One-Way," gives virtually continuous metal removal by having a series of broach carriers mounted on an endless chain. The complete machine is shown in Fig. 1.

The broach carriers move downward on the front of the machine. To ensure accuracy, they enter precision ways before they contact the work. The part being broached remains stationary, of course, in suitable holding fixtures on the work platen. After entering the ways, the broach carriers become individual rams. The chain applies only down-pull on the ram, and as with conventional surface broaching, gibs and ways support the ram while the broach teeth are cutting.

This chain broach principle provides greatly increased length of broaching stroke in a relatively small machine. The "One-Way" machine stands only 11 ft 6 in high, but has an effective broach length of 12 ft 10 in. The machine is equipped with a completely mechanical drive that includes a variable speed unit which gives stepless variation of cutting speed from 10 to 50 surface feet per minute. This variable speed feature allows the machine to be used at optimum cutting speeds for a wide variety of materials. Pulsations and vibrations sometimes incurred in other drive systems are eliminated by the mechanical drive.

The versatility of the machine is further increased by the wide face for mounting broaches on the chain-mounted rams and the wide work platen for taking indexing or stationary fixtures. These features are illustrated in Fig. 2. They allow the use of multiple tooling for simultaneous broaching of similar or different sides side by side. If the parts to be broached, require less than half the available stroke, 12 ft 10 in on the standard machine, duplicate tooling can be installed in sequence on broach carriers and the controls modified to provide two or more complete broaching cycles for each revolution of the chain. Similarly, with suitable broach spacing and control stops, two parts requiring a short stroke and one part requiring a long stroke can be broached side by side during a single revolution of the chain.

Other features of the machine are:—

(1) There is no return stroke of the broach. This materially reduces overall cycle time and increases the production rate.
(2) Tool life is increased owing to the smooth cutting action provided by the mechanical drive.
(3) Drive is by a conventional 25 h.p. alternating current motor and variable speed drive. A reversible motor is not required.
(4) Start and stop of drive motor and variable speed drive can be eliminated by the provision of an electric clutch in the drive mechanism.
(5) There is force-feed lubrication to all points of major wear.
(6) Electrical controls are at the extreme front of the machine. Safety locks are provided on the controls.
(7) A pit is not necessary; the operating platform is only 33 in high.

A specially designed floating "head" support for the upper idler sprocket automatically maintains correct chain tension and compensates for chain and sprocket wear. This reduces maintenance and eliminates down-time for adjustment. Of unusual interest is the fact that the tower of the machine is so constructed that it can be raised

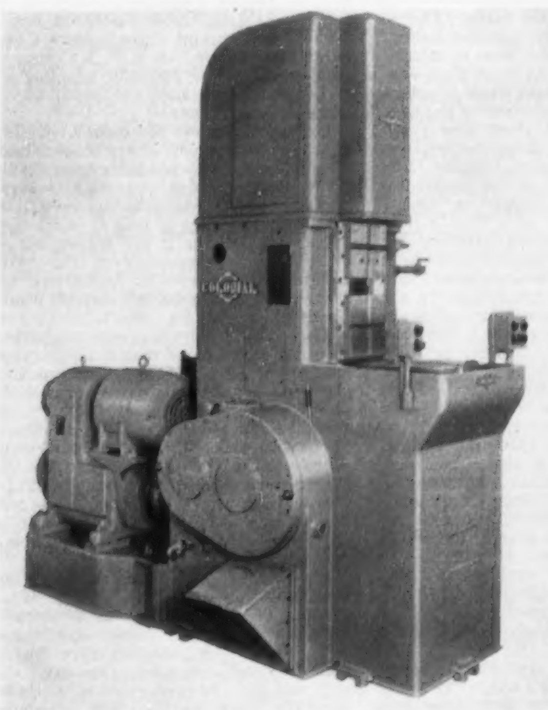


Fig. 1. Standard "One-Way" broaching machine. It provides an effective broach length of 12 ft 10 in

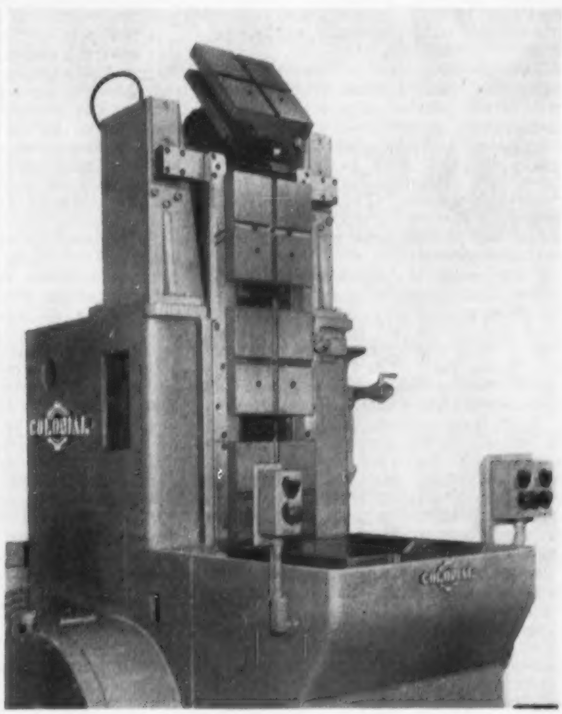


Fig. 2. Cover removed to show the wide faces for mounting broaches on chain-mounted rams

with extension inserts to allow a longer broach carrier chain to be installed. Additional carriers may then be attached to the chain to increase the effective broaching stroke from the standard 12 ft 10 in to as much as 18 ft. Actually, for each foot of spacer inserted in the column, two feet of stroke may be added.

The unusual versatility of this machine as to tooling, type of material to be broached and the form of the parts makes this machine of value for long, high production runs with complete automation; at the same time the machine is also suitable for broaching operations on relatively small batches of work.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Gas Turbines and Jet Propulsion (Sixth Edition).

By G. Geoffrey Smith, M.B.E., revised and enlarged by F. C. Sheffield.

London: Iliffe and Sons Ltd., Dorset House, Stamford Street, London, S.E.1. 1955. 8¾ × 5½. 412 pp. 342 illustrations and 4 fold out drawings. 35s. 0d. net.

Since it was first published as a small volume in 1942, "Gas Turbines and Jet Propulsion" has been recognized as the standard work in its own field. Edition by edition, the book has necessarily been enlarged, and this latest edition deals in a most comprehensive manner with the whole subject of the gas turbine as applied in aviation to-day.

The opening chapters follow closely on those in the immediately previous edition, but the later chapters, although generally following the same plan as the previous edition, have been completely rewritten to take full cognizance of the many developments that have taken place since the fifth edition was compiled. For example, combustion and fuels and fuel systems are given much more comprehensive treatment. Similarly, and understandably, ramjets, pulsejets, and rockets and rotating wing propulsion, are treated at much greater length.

The chapters on combustion fuel, systems and metallurgical problems, are particularly valuable. As is well known, development on these various aspects of gas turbines and jet engines is proceeding apace, and this work keeps abreast of these new developments in a truly remarkable manner. These chapters will be of great interest not only to the general reader but also to those engaged in this field of engineering.

Other valuable chapters deal with British and Canadian, American and European gas turbines. In these chapters, brief descriptions are given of the most important of current engines that are off the secret list. The lines on which development is taking place in the various countries is clearly shown.

Apart from aviation, the gas turbine is as yet still in the experimental stage as a power unit for other applications, but much interesting work is being carried out and encouraging results have been obtained in several fields. About 70 pages are given over to discussing and describing these developments.

As an introduction to the subject of gas turbines and jet propulsion, this work could scarcely be bettered. But it is considerably more than a mere introduction; it is, in fact, a conspectus of all the available information concerning developments in this fascinating field of engineering.

A review of this work would be incomplete without a reference to the very high standard of the illustrations. The half-tone and line illustrations are all, without exception, included to reinforce the exceptionally clear exposition of a difficult subject, which makes this a work that will repay close study both by amateurs of the subject and of those professionally engaged in connection with gas turbines.

The Design of Vehicle Transmissions (Gestaltung von Fahrzeuggetrieben).

By H. Reichenbächer.

Berlin: Springer-Verlag, Germany. 1955. 6 × 9. 154 pp. 164 illustrations. Price 26.40 DM.

This book is the fifteenth of a series on the design of machines and their components, published under the general editorship of Prof. K. Kollmann. These works are planned to provide assistance to designers and advanced students, and an outstanding feature of this and other works in the series is that, in presenting the relevant information, there is always a strong appreciation of the designer's point of view, problems and difficulties.

The volume under review should be of great value to automobile engineers. It presents a wealth of most useful information in a most lucid form and in a manner that permits a full appreciation of existing designs and an appreciation of their suitability for envisaged applications. Furthermore, it also fosters constructive thoughts and rational solutions about new problems in connection with mechanical and hydraulic transmissions for vehicles. Full praise must be given to the author for the production of an integrating volume of outstanding merit.

A general introduction outlines the scope of the book and discusses basic gear transmission forms, together with data concerning their efficiency. In the next chapter, 22 pages are devoted to the problems of gear ratio choice. Particular attention is paid to the manner in which acceleration requirements affect choice, and typical examples are worked out and analysed in some detail. It is rightly pointed out that the relevant data should be considered on the basis of the time-distance results, since these are more important than the more common considerations based on time-velocity data. To encourage this mode of approach, the author puts forward a graphic method of performance evaluation. Here, it may be pointed out that a very simple and effective method of dealing with this problem has been devised by W. Müller (*Eisenbahnanlagen und Fahrdynamik*, Vol. 2, 1953).

Gear transmissions and their development are dealt with in a chapter of 66 pp. One section is given to direct hand gear changing, its problems and design solutions. A second section deals with synchromesh gear designs, with particular reference to those due to Porsche, Z.F., Mercedes, Krupp and Mylius. The last-named, perhaps, deserved a more detailed reference because of its simple yet ingenious operating gear. Whilst on the subject of spur gears, the author also considers freewheel arrangements and semi-automatic transmissions of the type developed by Chrysler (M6), as well as Kreis and Ardelt. Transmissions with clutches for each gear are represented by Maybach, Minerva and Z.F.-Media designs. Two further sections deal with hand and foot, motor cycle transmission control.

Epicyclic gears are considered with reference to gear and torque ratios, and various types of clutch and brake arrangements, such as Wilson, Cotal, Dynaflow and Ford-Mercury. The chapter also includes a section on gear tooth design, which, because of the design data and worked out examples, is of great value. Two further sections dealing with bearing loads and shaft stresses conclude this chapter.

The next chapter (22 pp) deals with hydraulic couplings and their performance. Their application is illustrated with the help of a detailed description of the Hydra-Matic transmission. A comparative acceleration analysis for a 0·64t sports car, powered by a 25 h.p. engine fitted with either a hydraulic coupling or a two-speed mechanical transmission, concludes the chapter.

Torque converters are considered first in terms of their basic characteristics and then in terms of various designs. The effect of the torque converter is considered in relation to vehicle acceleration and climbing ability, as well as in relation to fuel consumption at full output and at part loads. A summary of the most important points to be considered in developing a satisfactory layout, concludes the book.

The bibliography is not altogether adequate. It is based mainly on German references. Apart from the improvement that could be effected by the inclusion of works published in other countries, it would be of greater value if the publishers of the quoted works were invariably stated. That the important papers on torque converters by Martyrer (Z.V.D.I.-1952), Hennings (A.T.Z.-1952) and Ziebart (Z.V.D.I.-1953) do not appear in the bibliography is a matter for some surprise.

In all, this is an excellent work on automotive transmissions. The clear analysis is a joy to read in connection with the problems of layout and vehicle performance. It can be very highly recommended. Needless to say, the drawings and the production of the book generally keep to the high standards expected of Springer publications. This work is not available in an English translation.

AUTOMATIC ANALYSIS OF NOISE AND VIBRATION

An Integrated Range of Danish Acoustic Equipment

THE complexity of modern industrial design demands that greater attention be paid to the reduction of noise and vibration. These effects may extend from a merely objectionable noise to a possibly self-destructive vibration that will cause a failure after some period of operation. Although a prototype unit of, say, an engine may be approved, the many component parts of which it is constructed, and the many variables encountered in operation, make it desirable that production line tests should also be carried out. These tests would indicate general noise levels, both as sound pressure levels and as acoustic levels, and would record the various individual frequencies of noise and their levels of intensity. Similarly, the various vibrations that occur can be recorded and analysed under various running and operating conditions. In addition to showing whether a component or a unit is operating within the prescribed limits, the analysis will usually indicate the frequency of a fault, which will often give a direct clue to the trouble.

Fig. 1. Automatic Spectrum Recorder, type 2311, comprising ⅓rd-octave Spectrometer and Level Recorder

Previous articles in *Automobile Engineer** have dealt with the principal design factors in noise reduction, and with test equipment concerned mainly with broadband spectrum analysis. To meet the requirements of both design and production analyses the next logical requirement is for a system that will:—

(a) Record directly noise via a microphone, vibration via an accelerometer and a pre-recorded signal via a tape recording.
(b) Record true sound pressure levels and acoustic levels via weighting networks.
(c) Analyse continuous and line spectra into the various frequency components on to a pre-printed chart for immediate and permanent use.
(d) Perform the analyses and recording automatically so as to eliminate point-to-point measurements and thus to increase efficiency.
(e) Perform automatic analysis over the whole frequency range, or a portion, at varying time cycles and life testing at set frequencies.

Equipment suitable for both research and production line techniques has been designed to international standards[1] by Brüel & Kjaer of Denmark. The basic instrument, the Automatic Spectrum Recorder shown in Fig. 1, was used to obtain the various spectrograms. The frequency/amplitude trace, Fig. 2, was recorded on the pre-calibrated rectilinear chart by the Level Recorder section driven by the signal derived from the ⅓rd-octave spectrometer. The noise from the source was picked up by a Condenser Microphone having a flat characteristic from 20 to 16,000 c/s $\pm$ 2 dB.

Vibration recordings, as in Fig. 3, are made in the same way except that a Barium Titanate Accelerometer is used. The first trace was recorded from the base of a motor mounting, while the second shows the reduction in vibration due to damping when measured on the immediate surrounding floor. The two superimposed traces, Fig. 3, are obtained by winding back the six inches of chart length and recording a second time without readjusting any controls.

The automatic system

A Wave Analyser based on either hetrodyne or degenerative principles, will usually provide a finer analysis of frequency but due to its design limitations will not lend itself to automatic sweeping of the full audio frequency range without band switching. The Brüel & Kjaer Spectrometer (the top half of the instrument in Fig. 1) is therefore a practical compromise and consists of 27 ⅓rd-octave filters and three weighting networks, fed by a pre-amplifier. A manual/automatic switch after the filters connects successively each of the 30 filters to the output amplifier which will drive either the moving-coil meter or the Level Recorder (the bottom half of instrument in Fig. 1). The linear stylus movement of the Recorder is governed by a driving coil in a magnet

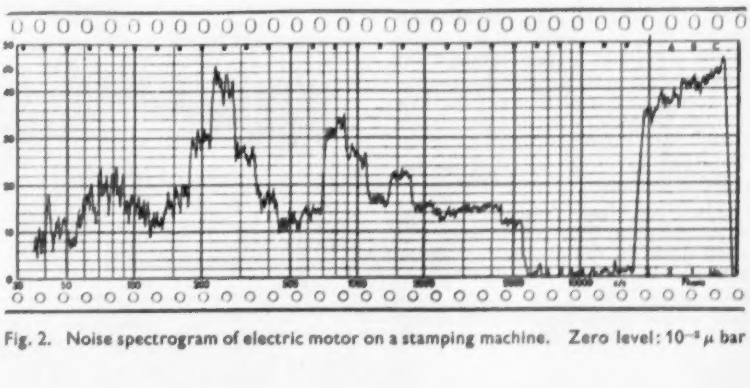


Fig. 2. Noise spectrogram of electric motor on a stamping machine. Zero level: $10^{-4}\,\mu$ bar

**November* 1951, *December* 1951, *February* 1952, *August* 1952, *July* 1954, *May* 1955.

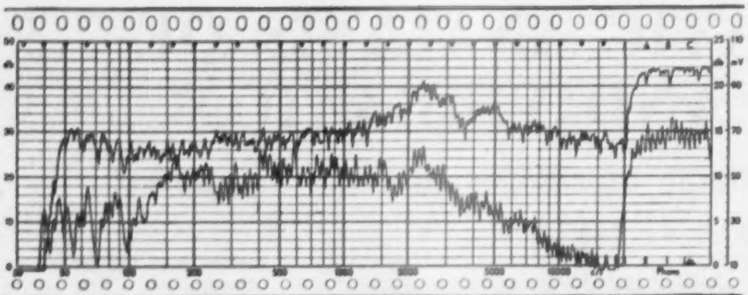


Fig. 3. Automatic recording of vibrations in and around a vibration-damped motor foundation

system. The coil position is determined by a D.C. signal derived from the input A.C. amplifier and detector circuits and the stylus, therefore, will respond to the change in the level of a signal and not its frequency variation. The automatic switching of the 30 filters in the spectrometer is done by a mechanical linkage from the Level Recorder gearbox. Chart speeds ranging from 10 cm/sec to 1·08 cm/hr can be selected according to the method of test, and the gearing is arranged so that the spectrometer frequency characteristic is synchronized with the preprinted frequency chart. Alternatively, the spectrometer can be driven through its range at a very slow speed relative to a plain or preprinted recording chart so that any one filter will be in operation for some minutes.

Methods of analysis

The form of the noise or vibration and the type of measurement required will decide the method to be used. In automobile engineering it is very rare to find a noise consisting of a single tone with, perhaps, harmonics. In fact, due to the many constructional parts each having a natural frequency of vibration, the practical noise wave-form will be extremely complex. It will probably contain a number of major frequencies and their harmonics plus other frequency components, all of various amplitude levels. If all of the frequency components are occurring continuously it is a simple matter to obtain a spectrogram. The main factor is to ensure that the spectrometer remains on each filter position long enough for each frequency component within that ⅓rd-octave band to occur. This can easily be adjusted by the gearbox settings.

There are often occasions, however, when the frequency components of a noise signal are not constant. A typical example of this is the starting and acceleration run of an engine when both frequencies and amplitudes change during the time. The requirement might be to record the noise level during, for example, a 20 seconds run, or perhaps to examine the vibration analysis of certain structural members,

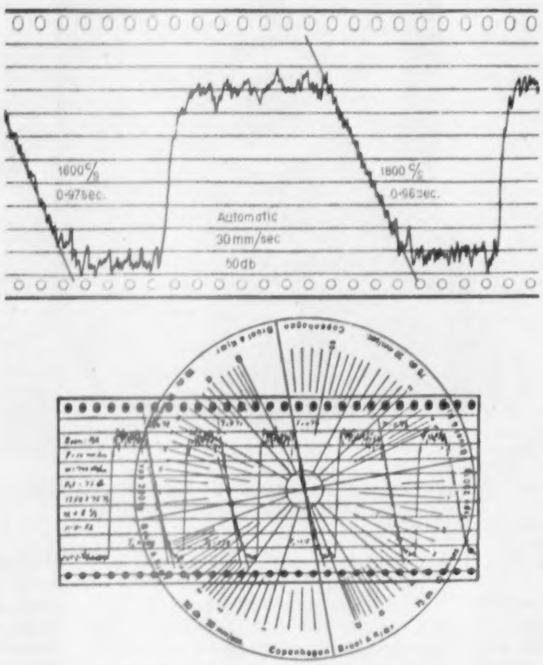
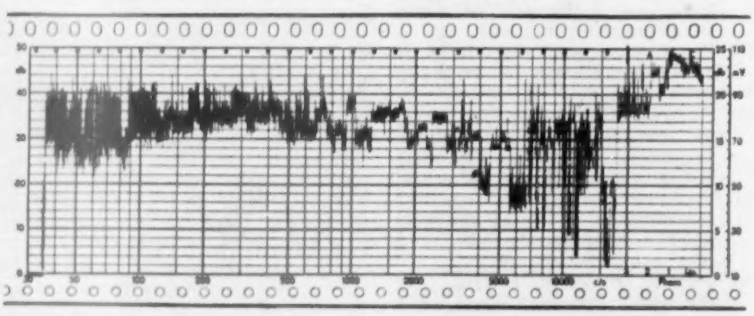


Fig. 5. Measurement of reverberation time

during acceleration. For the noise level measurement a paper speed of, say, 10 mm/sec could be chosen and plain paper used, bearing in mind that one centimetre relates to one second on the time scale. The spectrometer can then be set permanently to any of the three phon filters, or to the linear amplifier position, and a noise level record obtained. To obtain an analysis of all the vibrations (or noise) during this 20 seconds run is more complicated, however, because it is necessary to analyse amplitudes as a function of time and frequency. The method now being used is to make a tape recording of the vibration to be analysed and to form this into a tape loop with a mark made by a point magnet at the start. The gearing of the Automatic Spectrum Recorder is then adjusted so that for every full rotation of the tape the spectrometer remains on one filter position, while the level recorder chart is running at 1 cm/sec. When the tape loop has rotated 27 times the Spectrometer will have analysed the tape signal 27 times—once for each ⅓rd-octave position—and the chart will show 27 successive recordings. These records can then be laid out with their commencements vertically in line and the various frequencies and amplitudes of vibration can be read off against a common time scale. This method can also be adapted to the measurement of transient sounds of relatively short duration.

Long term measurements

During some tests it is possible that there may be some uncontrolled factor which will cause errors in readings. This could occur when taking a cabin noise analysis during a road trial, where road surface, wind, speed, etc. cannot be held constant. In such a case the spectrograms and noise levels are recorded over a relatively long period of time. An example of this technique is shown in Fig. 4, where the spectrometer is allowed to stay on each filter for sometime. The complete chart, in fact, took 20 minutes to record. In this way, both the maximum and minimum levels can be determined for the various frequencies and a practical average obtained.



Potentiometer 75 dB. Paper speed 0·3 mm/sec

Fig. 4. Long-time recording of spectrograms and noise levels

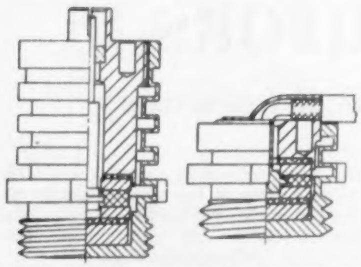


Fig. 6. The new barium titanate accelerometers, types 4306 and 4307

Reverberation measurements

In addition to direct measurements of noise, investigations are made into the efficiency of sound proofing materials and of the interior structural design of cars in order to reduce the reverberation time. The reverberation time is the period required by the sound energy to fall by 60 dB. These measurements can easily be made by using a special protractor to read off the reverberation time from the decay curve on the Level Recorder chart. Fig. 5 shows, in the centre of the chart, the decay curve of a signal of 1,800 c/s. The lower diagram shows the protractor superimposed on a chart recording in the way that it is used for measurements.

Microphone and accelerometers

When accurate recording equipment is used, the various accessories must also be of a high quality and, for use with the Automatic Spectrum Recorder, a Condenser Microphone, type 4111, has been designed. Combining small dimensions with high stability and having a cathode follower tube in the head, the microphone derives its polarizing and heater voltages via the mounting socket on the front panel of the spectrometer. Having a sensitivity of 2mV/μbar, the distortion is less than 4 per cent up to 140 dB over $2 \times 10^{-4}\mu$bar.

For vibration measurements one of the latest instruments uses a ceramic-barium titanate accelerometer which, when suitably prepared and polarized, gives a uniformly high piezo-electric effect with large capacity and good stability. The design of these, shown in Fig. 6, is such that the outer casing acts both as a protective body and as a spring providing the essential static pressure on the ceramic discs.[2] Since these accelerometers have been designed only recently, the accompanying table of characteristics will be of interest.

CHARACTERISTICS OF BARIUM TITANATE ACCELEROMETERS

Accelerometer type	4306	4307
Sensitivity in mV/g (g=acceleration of gravity)	20 approx	6 approx
Capacity in pF	900	900
Weight in grams	16	7
Resonance frequency in kc/s	25—30 approx	45 approx
Usable frequency range in c/s	2—15,000—20,000	2—28,000
Max. acceleration to be allowed, multiples of g (981 cm/sec^2)	2,000	5,000
Smallest leakage resistance in ohms	10^{11}	10^{11}
Smallest acceleration to be measured with the combination Pre-amplifier 1606—Spectrometer 2109 (or Analyser 2105), cm/sec^2	0·25	1·00
Damping for transverse vibrations, dB	25 minimum	30 minimum
Greatest diameter, mm	15	15
Height, mm	20	10

Constant sound pressure source

A new development, just released by Brüel & Kjaer, is the model 4211. It is designed as a special sound source for testing purposes where a high sound-pressure level is required at the entrance of the unit to be tested, such as a carburettor inlet, exhaust pipe, silencer, etc. In these cases a well-defined and constant sound pressure at different frequencies is applied to the entrance of the item under test, while perfect sound insulation is required for any sound path other than the one from the sound source through the test piece. The Constant Sound Pressure Source, with the test object mounted on top of it, should be placed in a hard-walled chamber where a microphone registers the sound level of the diffused sound fields which are propagated by the test object. Backward radiated energy from the sound source is sufficiently absorbed by a terminal layer of Rockwool, 40 cm thick. The method used is shown in Fig. 7, where item 2314 is a Beat Frequency Oscillator (B.F.O.) driving the sound source 4211, and items 4111(1) and 2601 are a microphone and amplifier feeding back to the B.F.O. compressor in order to maintain a constant level of sound output. The diffused signal registered by the Condenser Microphone 4111(4) is then analysed by the $\frac{1}{3}$rd-octave Spectrometer 2109 whose output goes to the Level Recorder (bottom half of item 2314). Here the chart is synchronized with the B.F.O. frequency so that the recording shows the diffused sound level at the various frequencies.

Setting datum levels

A practical problem often encountered in vibration measurement is to set the overall sensitivity of equipment to a known datum. The Vibration Pick-up Pre-amplifier 1606 incorporates an electro-mechanical oscillating disc which enables pick-ups to be calibrated. The output can then be used as a reference level of acceleration, velocity or displacement. Suitable integrating networks are provided so that a, v, and d signals are available without readjustment of any controls.

Acknowledgments are made to Dr. Per V. Brüel, of Brüel & Kjaer, Naerum, Denmark, and J. Foreman, Assoc. Brit. I.R.E., of Rocke International Ltd., 59 Union Street, London, S.E.1, the U.K. representatives.

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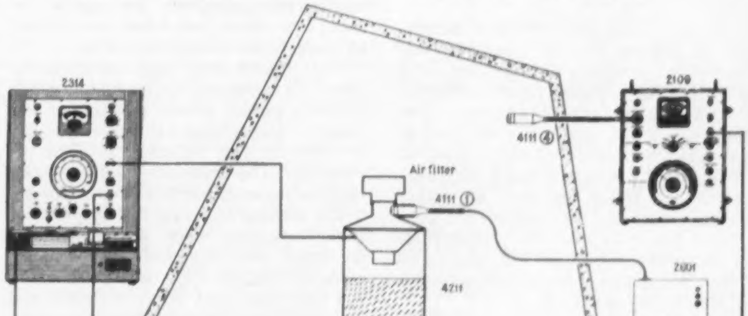

Fig. 7. Scheme for the determination of sound attenuation through an air filter with the aid of the Constant Sound Pressure Source, type 4211

RED REAR REFLECTORS

Testing Techniques at the National Physical Laboratory

ON 1st October, 1954, a law came into force which made it compulsory for red rear reflectors to be fitted to all road vehicles. Although a red rear reflector is of quite simple construction, its effectiveness depends on a number of factors. Very simple tests have been devised for examining the intrinsic quality of reflectors and determining the cause of poor performance.

For many years the quality of reflectors fitted to British cars has been controlled by regulations issued by the Ministry of Transport. The British Standards Institution has also recently drawn up specifications for red rear reflectors, which were published last year as B.S.2515:1954. Manufacturers are under no legal compulsion to submit reflectors for testing, but a certificate for a successful test greatly enhances the sales value of the product and is essential in many overseas markets.

Testing is carried out at the National Physical Laboratory, Teddington, Middlesex. This work used to be undertaken by the Light Division of the Laboratory, but in 1954 it was transferred to the Test House, which was established a few years ago to relieve the research divisions from routine testing.

Depending on the market in which they are interested, manufacturers may submit consignments for one of two different tests, a consignment being a representative sample of the many thousands manufactured from one series of moulds. One test consists of a photometric examination of a reflector over the angles of view and entrance laid down in the Ministry of Transport Regulations. The other is a test for compliance with B.S.2515, which includes, in addition to the photometric measurements, a colour measurement and some mechanical tests.

In view of the importance of achieving the highest possible quality, it was considered desirable to make provision in the British Standard for two different grades of red rear reflectors. The specification for Grade 1 is intended as a standard to be achieved where possible, while Grade 2 is regarded as a minimum requisite. When testing to either specification, the examiners are concerned not only with the performance of complete reflectors but also with individual components and the materials of which they are made.

The photometric examination of red rear reflectors is carried out in a range over 100 ft. long, which was specially constructed for the job. The Ministry of Transport Regulations stipulate that when a reflector is illuminated by a beam from a projector of 2,000 candelas 100 ft away, the minimum reflected intensity of the red light shall be not less than one-thousandth of a candela.

The instrument used for photometric measurement is a visual telephotometer, designed and built in the Light Division of the National Physical Laboratory, which has been specially adapted for this type of work by the inclusion of an illuminating system. This consists of a ribbon filament lamp focused on an aperture, an image of which is projected to a distance of about 100 ft to form the illumination of the reflector. The same lamp conveniently serves as the source of illumination for the comparison field of the telephotometer. The lamp illuminates a mirror, the position of which can be adjusted to vary the angle at which the beam is projected relative to the receiving position. The light reflected back to the instrument is viewed through an eyepiece.

The apparatus is calibrated by substituting a magnesium oxide disc of known reflecting power for the sample to be tested. A photometric balance is then made with a red filter of a known transmission placed across the test and comparison fields. The subsequent photometric measurements on a reflector are made using the red filter on the comparison side of the field only. Thus all the data necessary for evaluating the effective reflection factors of the reflector are known. The ribbon filament lamp is so run that the light falling on the reflector approximates to that of a motor car headlamp operated at a colour temperature of 2,854 degrees Kelvin.

The British Standard further provides that the colour of the red light shall be in accordance with the specification for Signal Red, Class A, laid down in B.S.1376, which calls for a blue content of not more than 0·020 and a green content not greater than 0·335. The colour measurements are made by using a modified Donaldson colorimeter in place of the comparison field.

Any uncoloured light after being observed is blocked out by a small mask of black paper, placed in front of the reflector and arranged in size and shape to obscure the reflector as little as possible. The remaining red light can then be measured with the colorimeter.

On looking through the eyepiece of the photometer when the colorimeter is fitted, the observer sees a split circular field of view, one field being red light sent back by the reflector and the other the light from the colorimeter. By adjusting three knobs, each controlling a separate colour, the colour of the field formed by the colorimeter is altered until it exactly matches that of the reflector. Three scale readings are taken which (after the instrument has been calibrated using a standard white surface in the manner similar to that previously described), can be interpreted as the chromaticity coordinates of the reflected light.

For all these tests the reflector is mounted on a tilting support, so that its centre remains in a constant position, the required angles of view being obtained by tilting the reflector and altering the direction of incidence.

Four different types of mechanical tests are specified by the British Standard. Reflectors must be capable of withstanding the vibration of the car for an indefinite period. Their ability to do so is examined in a machine of special construction, which is vibrated at approximately 750 cycles per minute through a distance of ¼ in. This machine consists essentially of a beam spring-mounted at one end, which is under a spring tension of between 60 and 70 lb. An electrically driven cam lifts the beam and causes it to fall very sharply on to a steel anvil, thus producing a heavy impact once during each revolution. The reflector is mounted in its normal operating position and subjected to this treatment for at least an hour. At the end of this period there must be no evidence of any material weakness or displacement of parts.

Another test is for penetration of moisture. In this test the reflector is mounted in the normal operating position and a solid cone spray is projected on to it at an angle of 45 degrees, the precipitate amounting to 0·1 in per minute. The reflector has all drain holes open and is revolved about its vertical axis at the rate of 4 r.p.m. for at least 12 hours. It must then show no visible retention of moisture.

Resistance to corrosion is tested by subjecting a sample unit to a salt spray for a minimum period of 50 hours consisting of two periods of 24 hours' exposure, each followed by a drying time of one hour. On the completion of this test the reflector must be substantially free from corrosion.

The fourth test is for resistance to petrol. A sample component is immersed in petrol for a period of one hour, removed, and allowed to dry for one hour. The sample must not show signs of attack.

After a sample has passed the tests specified either by the Ministry of Transport Regulations or by the British Standard, the manufacturer is given a certificate and the reflectors are engraved with the N.P.L. monogram and a number. The value of these tests as a contribution towards the safety of our roads at night is indicated by the really high performance of some of the reflectors submitted for examination.

BLENDED FUEL EXHAUST GAS

Determinations with the Thermal Conductivity Analyser

Felix Gutman and K. Weiss*

THE establishment and means of adjustment of the air-fuel ratio are of vital importance for the satisfactory operation of petrol engines. Complete stoichiometric combustion of straight hydrocarbon fuels corresponds to a theoretical air-fuel ratio of 14·3:1; below this ratio, in the region of rich mixtures, combustion is incomplete due to lack of oxygen: unburned fuel, mainly hydrogen and carbon monoxide, appears in the exhaust gas. In the lean region, at air-fuel ratios in excess of 14·3:1, combustion is complete: no hydrogen, very little methane, only very small amounts of CO plus a decreasing concentration of CO_2 appear in the exhaust gas, in addition to unconsumed oxygen.[1]

The analysis of exhaust gases by means of the "thermal conductivity gas analyser" is a well established method.[2] It is based upon the fact that the thermal conductivity of hydrogen is about 7 times that of nitrogen or oxygen, while the thermal conductivity of CO_2 is about two-thirds that of N_2. Thus, even traces of H_2 or CO_2 in air are readily measurable. The apparatus[3] consists, in principle, of a platinum filament which is electrically heated at a constant rate and exposed to the dry gas in a closed chamber. The filament eventually attains an equilibrium temperature dependent on the rate at which heat is applied to and removed from it; since the heat input is kept constant its temperature depends only on the rate of heat loss, the gas being assumed to be at ambient temperature. If heat is removed from the filament purely by convection, then the equilibrium temperature will depend only on the thermal conductivity of the gas. Since the electrical resistance of the filament is very nearly a linear function of its temperature, a comparison of the resistance of the filament when surrounded by exhaust gas or by a standard gas (usually air) yields a means, through an empirical calibration, of estimating the thermal conductivity of the exhaust gas.

It is well known[1,4] that for straight hydrocarbon fuels, the air-fuel ratio may be uniquely determined by the estimation of either H_2 or CO in the rich range and by that of CO_2 or O_2 in the lean range. Thus the electrical thermal conductivity gas analyser represents a convenient means of measuring the air-fuel ratio and large numbers of such instruments are in use.

It was the purpose of this work to investigate the response of the thermal conductivity analyser to blended fuels used in high performance engines and to study the performance of the method.

Experimental equipment

The measurements were carried out with the aid of a Ricardo E6, single cylinder, 4-stroke variable compression engine, coupled to a swinging field electrical dynamometer. The carburettor was a Solex type F.A.1 fitted with a 27 mm choke. A variable main jet allowed variation of the fuel mixture during tests by means of a tapered needle valve. A 2-gallon tank as main supply, a small sampling tank and a calibrated measuring burette comprised the fuel measuring apparatus. The time for 50 c.c. of fuel to be burned was then measured with a stop-watch, while the air consumption was determined by means of an "Alcock" viscous flow air meter. The engine speed was kept constant at 2,750 r.p.m.

Readings were taken of barometric pressure and wet and dry bulb temperatures. Air-fuel ratios were calculated from the observed values for fuel and air consumed, and corrected for humidity. Since water vapour has a thermal conductivity of about three-quarters of that of air, a water trap was employed to condense and thus remove it from the exhaust gas, which was allowed to cool to ambient temperature.

The exhaust gases were chemically analysed by means of an Orsat gas analysis apparatus which received its samples immediately after the electrical analyser. These analyses were used as a check on the air-fuel ratios indicated, employing the relationship between this quantity and chemical composition given by D'Alleva and Lovell[1] and others[4].

Four analysers of three different designs were used, the circuit in all cases being a constant current Wheatstone bridge. A preliminary setting standardizes the bridge current and thus the filament equilibrium temperature when surrounded by air. Platinum filaments of 9 ohm cold resistance operated at 100 mA current with a 400 microamperes full scale deflection meter of 30 ohm internal resistance produced the best results; Pt-Ir filaments of 17·1 ohm cold resistance operated at 150 mA bridge current with a 500 μA meter of 50 ohm resistance gave only slightly inferior results. The operating temperature of the Pt elements (100 mA bridge current) was calculated to be about 60 deg C above ambient temperature and that of the Pt-Ir filaments (150 mA) about 70 deg C above ambient temperature. An attempt was made to raise the sensitivity of one of the instruments by increasing the bridge current and thus the filament temperature. This, however, rendered the instrument sensitive to the velocity of flow of the gas and introduced another element of non-linearity between its readings and the air-fuel ratio. This is shown in Fig. 1, which represents the calibration curves of two identical analysers, one operating with 150 mA (Curve 2) and the other with 200 mA bridge current (Curve 1).

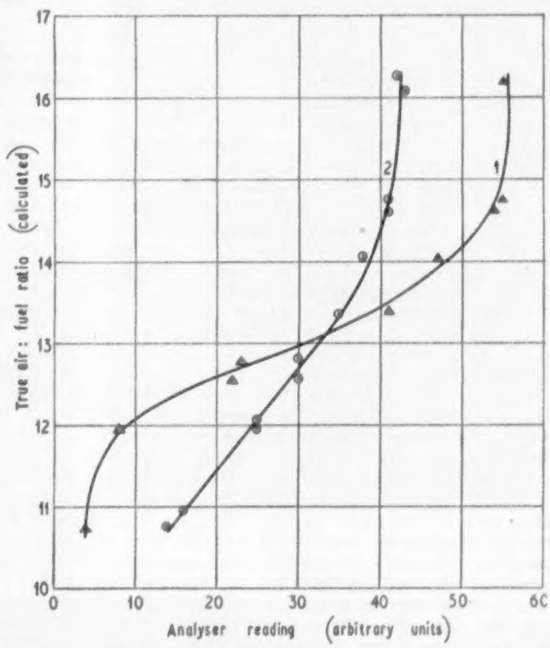

Fig. 1. Calibration of two thermal conductivity type gas analysers on straight hydrocarbon commercial petrols. Curve 1 refers to an analyser operating at 200 mA bridge current, and Curve 2 to an identical analyser with 150 mA bridge current

The maximum permissible bridge

*The New South Wales University of Technology, Sydney, Australia.

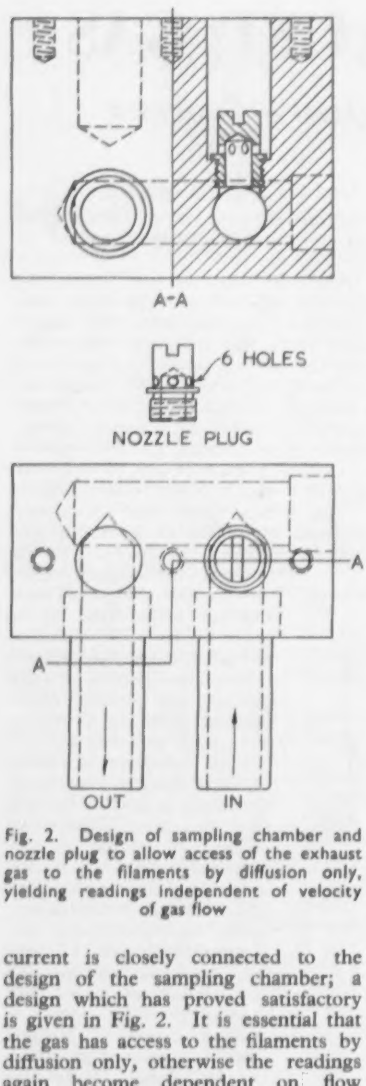

Fig. 2. Design of sampling chamber and nozzle plug to allow access of the exhaust gas to the filaments by diffusion only, yielding readings independent of velocity of gas flow

current is closely connected to the design of the sampling chamber; a design which has proved satisfactory is given in Fig. 2. It is essential that the gas has access to the filaments by diffusion only, otherwise the readings again become dependent on flow velocity; a special nozzle plug, of 8 mm diameter and screwed into the inlet port of the sampling chamber, was found to give satisfactory operation.

The location of the sampling tube within the exhaust gas stream is somewhat critical: if, for example, the sampling tube is located in a direction perpendicular to the direction of gas flow, the resulting disturbance of the main flow of gas tends to produce a partial vacuum in the sampling chamber, resulting in quite erroneous readings. However, if the sampling tube is mounted coaxial with the exhaust pipe with its opening facing up-stream, then a small positive pressure is produced which yields consistent, accurate and well-reproducible results.

Air-fuel Ratio for Maximum Power Output at 2,750 r.p.m. for various kinds of fuel

Fuel	A-F Ratio	Compression Ratio
Petrol "A"	12·25:1	7:1
Petrol "B"	13·13:1	7:1
Petrol "C"	13·38:1	7:1
Fuel "A"	9·71:1	11:1
Fuel "B"	8·11:1	11:1
Fuel "C"	10·6:1	9·5:1
Fuel "D"	7·54:1	12:1
	6·93:1	11:1
Fuel "E"	5·2:1	11:1
Fuel "F"	10·8:1	12:1
	7·6:1	10·5:1

Results

The following fuels were tested:

Fuel A, containing:

50 per cent ethanol,
25 per cent benzol,
25 per cent octane,

to which were added
6·5 per cent nitrobenzene, producing a fuel of specific gravity of 0·826.

Fuel B, containing:

60 per cent methanol,
20 per cent octane,
20 per cent benzol,

and having a specific gravity of 0·789.

Fuel C: Shell Racing Fuel Type "X."

Fuel D: Shell Racing Fuel Type 8/11, spec. gr. 0·790.

Fuel E: Shell Racing Fuel Type A, spec. gr. 0·792.

Fuel F: A mixture of equal parts of Shell Racing Fuels Types TT and M.

Tests were made also with two brands of straight, hydrocarbon, commercial, petrol of specific gravity 0·715 and 0·720, respectively. Plots of measurements on these two hydrocarbon fuels, using a compression ratio of 7:1, are shown in Fig. 1. Curve 2 refers to correct operating conditions. The abscissæ give analyser readings in arbitrary units on a 100 division scale. A single curve is seen to fit the readings for both petrols.

The response of the instrument is almost entirely determined by the hydrogen concentration within the rich range and by that of CO_2 in the lean region. Both responses are approximately linear, but have different slopes. If thus the air-fuel ratio is slowly increased towards 14·3:1, the ratio of complete combustion, the response of the analyser becomes non-linear in the vicinity of this value. Curve 2, it will be seen, follows a fairly good straight line in the rich mixture range, but its curvature becomes appreciable in the lean region.

A kind of saturation effect is evident at higher values of air-fuel ratio, and the readings virtually cease to increase beyond 14·8. This is probably caused by the rapid decrease in CO_2 concentration at air-fuel ratios above the stoichiometric ratio, and to the higher concentrations of oxygen then present: the thermal conductivity of the exhaust gas then tends to approach that of air. This effect is present with all analysers tested. It should therefore be stressed that this type of exhaust gas analyser yields results which are far less accurate for lean than for rich mixtures; in fact, the instrument can hardly be used at mixtures leaner than about 14·5:1. The analyser readings are practically independent of the compression ratio between 7:1 and 12:1.

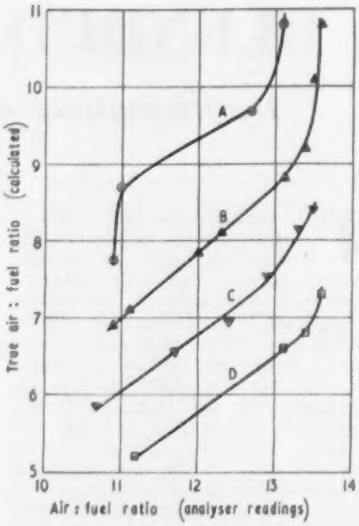

Fig. 3. Analyser readings obtained with 4 different kinds of blended fuels

Fig. 3 shows typical calibration curves for some of the blended fuel mixtures. The abscissæ are taken from the analyser scales which previously had been accurately calibrated and checked against values obtained with straight hydrocarbon fuels (commercial petrols). It will be seen that very large deviations are observed for all the blended fuels tested, the analysers always reading far too lean. This is due to the fact that these blended fuels contain large proportions of oxygen in their constituents, and therefore require less atmospheric oxygen for complete combustion. Thus fuel "B," containing methanol which has a stoichiometric air-fuel ratio of 6·44:1, produces a much larger shift in the analyser readings than, for example, fuel "A," containing ethanol requiring an air-fuel ratio of 8·84:1 for complete combustion. That the curves are rather more non-linear than those obtained with straight hydrocarbons, is probably due to the complexity of the combustion products which contribute to the thermal conductivity of the exhaust gases. The curve for fuel "A," containing nitrobenzene, shows the greatest curvature.

The average slope changes but little from fuel to fuel, but the whole curves are shifted towards lower values of calculated air-fuel ratio, very nearly parallel to themselves. Such a behaviour would be expected if it is the availability of chemically bound O_2 from within the fuel itself which is responsible for these changes. There is no correlation between the analyser readings and the specific gravity of the fuel. There is evidence of a "saturation effect" in the lean region similar to that observed with straight hydrocarbon fuel. The blended fuels permit the engine to run on much richer mixtures

than does straight petrol, and again the analyser readings are far more accurate in the rich than in the lean region.

It is of interest to compare the air-fuel ratio at which the fuels used deliver maximum power output: it is seen from the accompanying table that appreciable differences exist even between commercially available straight petrols. The air-fuel ratio for maximum power also depends on the compression ratio. Fuel "F," for example, yields maximum power at a/f ratio 10·8:1 at a compression ratio of 12:1, compared to 7·6:1 at a compression of 10·5:1; fuel "D" likewise produces maximum power output at an air-fuel ratio of 7·54:1 at a compression of 12:1, compared to 6·93:1 at a compression of 10·5:1. A higher compression ratio always produces maximum power with a leaner mixture.

The authors are indebted to Mr. E. C. Martin of the Analytical Department of the New South Wales University of Technology, for assistance with the Orsat apparatus, and to Mr. A. Vainomae who helped run the test engine.

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AUTOMATIC CYLINDRICAL GRINDING

An Adaptation of the Newall "LA" Machine

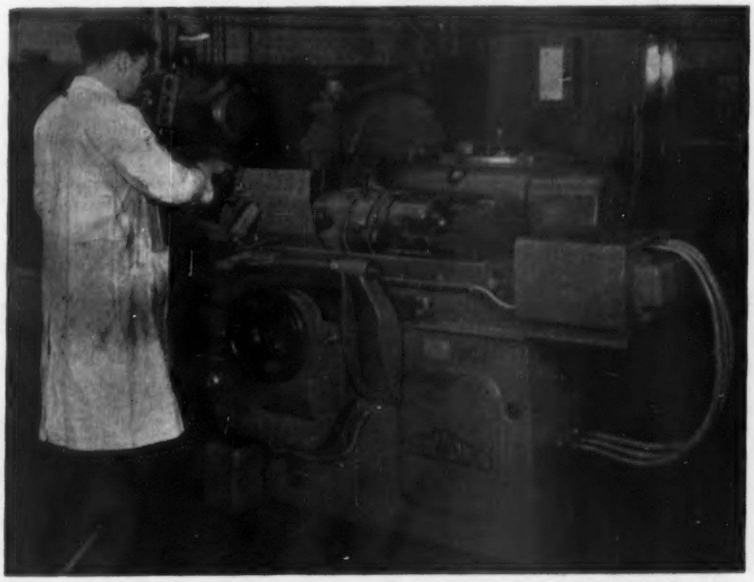
An interesting adaptation to give fully automatic operation on their "LA" cylindrical grinding machine has recently been introduced by The Newall Engineering Company Ltd., Peterborough. Essentially, the automatic features of the machine are:—

(1) Hydraulically actuated hopper feed.

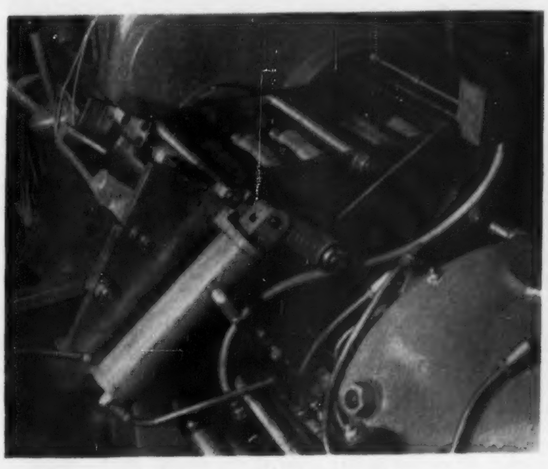
(2) Fully automatic grinding cycle controlled by a Chamberlain and Hookham timing unit.

(3) Air chuck for quick release and return of the tailstock centre.

As the hopper is filled, the first component is automatically positioned between the dead-centre workhead and tailstock centres, which close through operation of the air chuck. Rapid approach of the wheelhead follows and the workpiece commences to rotate. Through the nine-speed gearbox, workspeeds from 25 to 260 r.p.m. are available. On completion of the rapid approach of the wheelhead, the automatic diameter feed takes over, and when this reaches a dead stop there is a dwell period determined by the setting of the timing unit.



Newall "LA" machine adapted for automatic operation



Hydraulically operated hopper feed

At the end of the dwell period, the wheelhead automatically withdraws together with the diameter feed, rotation of the workpiece ceases, the tailstock centre is withdrawn through operation of the air chuck and the finished component falls into the lower receptacle of the hopper. A new workpiece immediately moves into position and the cycle is repeated.

A Newall table-mounted trumatic wheel-forming attachment is positioned to the right of the tailstock so that no dismantling is necessary for wheel dressing. When dressing is required, the table is moved to the left in correct relationship to the wheel. The wheelhead is set by the hydraulic system to a positive stop at its withdrawn position. This, in conjunction with the micrometer adjusted diamond dresser, allows the wheel to be dressed and grinding to be continued without there being the necessity of resetting the feed rate dial for sizing.

A new development incorporated in the machine is a hydraulic oil temperature control unit designed to eliminate any temperature rise above ambient. The importance of this feature is that a constant hydraulic cycle is maintained, thereby obviating that possibility of variation in component size that is normally associated with temperature fluctuations.

ELECTRO-DEPOSITION

Production Developments by Fescol Ltd.

TO cope with the increasing demand for their services as specialists in the electro-deposition of metals, Fescol Ltd., North Road, London, N.7, have opened a new factory at Leeds Road, Huddersfield. The new factory, part of which is shown in the accompanying illustration, has a floor area of nearly 14,000 ft^2 and is approximately twice the size of the factory hitherto occupied in Huddersfield. The production area now occupies over 10,000 ft^2, and includes a chromium deposition shop, a nickel deposition shop, stores and inspection departments.

There are six vats in the chromium deposition shop. They are installed in special pits to give a common working height. Above them is an electric hoist of one-ton capacity and several smaller hoists. Shafts up to 8 ft in length can be treated, but 4 ft 6 in is the maximum length that can be treated in one operation. Each vat can deal with a superficial area of 6 square feet at one time.

Power for each vat is supplied by an independent oil-immersed "Westalite" rectifier set. Five of the sets have an output of 1,500 amps at 12 volts each, while the sixth has an output of 1,000 amps at 12 volts. Each rectifier set is a self-contained unit, with mains and auto transformers and control gear. Fumes from the vats are extracted through ducting by high speed fans, and to ensure an adequate replacement supply of fresh air, four louvred inlet vents with sliding fronts are provided in the walls. These vents are so positioned that the cool air entering the building has first to pass and cool the transformers and rectifiers adjacent to the vats.

The vats for the deposition of nickel are also installed in pits to give a common working height. These vats are served by a main electric hoist of 2 ton capacity. They will accommodate the same size shafts as the chromium deposition vats. The power supply is obtained through motor generator sets having a total output of 3,000 amps at 10 volts. In case of power supply failure, accumulators with a capacity of 4,000 amps at 10 hour rate discharge have been installed.

A recent Fescol development is a process for the direct deposition of chromium on light alloys. One of the major problems formerly encountered in the production of electro-depositions for engineering purposes was to obtain adequate adhesion on light alloys. After intensive research, Fescol Ltd., solved the problem in some degree by evolving a process which enabled chromium to be deposited on a limited range of light alloys.

After further investigations, an improved process has been developed. It enables the company to deposit chromium direct on to all known light alloys in such a manner that the deposited metal becomes an integral part of the base metal. The new process has received Ministry of Supply approval for use on aluminium and aluminium alloys used in the construction of aircraft.

The Fescol process differs from many other commercial methods in that the chromium is deposited directly on to the light alloy and does not rely on an undercoat of a second metal to ensure adhesion. Special plant for carrying out this process has been installed in the new Huddersfield factory.

The new Huddersfield factory of Fescol Ltd., showing (left) the chromium deposition tanks; (centre) the control block—inspection stores, etc.—and (background) the nickel deposition tanks

THE AUSTIN CHAMP

A Vehicle That Has Exceptionally Good Cross-Country Performance

Part II

Continued from page 362

A BORG and Beck single dry plate clutch is employed. It is 10 in diameter and has 39 in^2 friction lining area. The twelve pressure springs, when assembled into the unit, exert a load of 1,440-1,560 lb. A ball thrust bearing is employed in the withdrawal mechanism.

The dry weight of the gearbox is 103 lb. All five forward speeds are fitted with synchro-units. The reverse gear train is not in the gearbox, but is housed, together with the transfer gear, in an extension of the rear axle casing. This arrangement has been adopted to give the same number of ratios in reverse as in forward drive. It also reduces the width of the gearbox so that there is more space available for the driver and front passenger than would be possible with the more conventional layout. The overall ratios are: top 4·98:1, fourth 7·62:1, third 11·84:1, second 17·71:1, first 27·25:1, and reverse 26·53:1. They include the final drive ratio of 7:34 and the transfer gear ratio of 1·02:1.

A D.T.D.133C aluminium alloy casing is employed. It is fitted with a Tecalemit breather, which is so designed as to prevent the entry of water if the box is cooled suddenly when the vehicle is wading. The ball bearing carrying the primary shaft, together with the flanged, cupped housing for this bearing, is in an aperture in the front wall of the casing. A counterbore is machined round the aperture, to receive the flange of the bearing housing. This flange is clamped in the counterbore by the D.T.D.300 die-cast aluminium alloy front cover, which also retains the outer race of the bearing in the housing. The lower portion of the housing is machined to clear the layshaft gear; a dowel in the flange and the front wall of the gearbox provides location against rotation. The inside diameter of the bearing is approximately $1\frac{3}{8}$ in and the outside diameter is slightly more than $3\frac{1}{2}$ in. A ring nut on a $1\frac{7}{16}$ in diameter threaded portion of the primary shaft retains the inner race of the bearing against the primary gear. This nut is locked by a tab washer. Forward of the nut is a lip type oil seal that bears on the shaft and is housed in the front cover. The thrust bearing of the withdrawal mechanism of the clutch is carried on a tubular extension of the front cover. Inside the extension, the diameter of the primary shaft is about 1 in.

A phosphor bronze bush in the tail end of the crankshaft carries the front end of the primary shaft. Immediately behind this, the shaft is increased to about $1\frac{1}{4}$ in diameter and splined to carry the centre component of the clutch. The tooth ring and cone for the synchro-mechanism for top speed are machined on an up-ended portion at the rear end of the shaft. The

SPECIFICATION

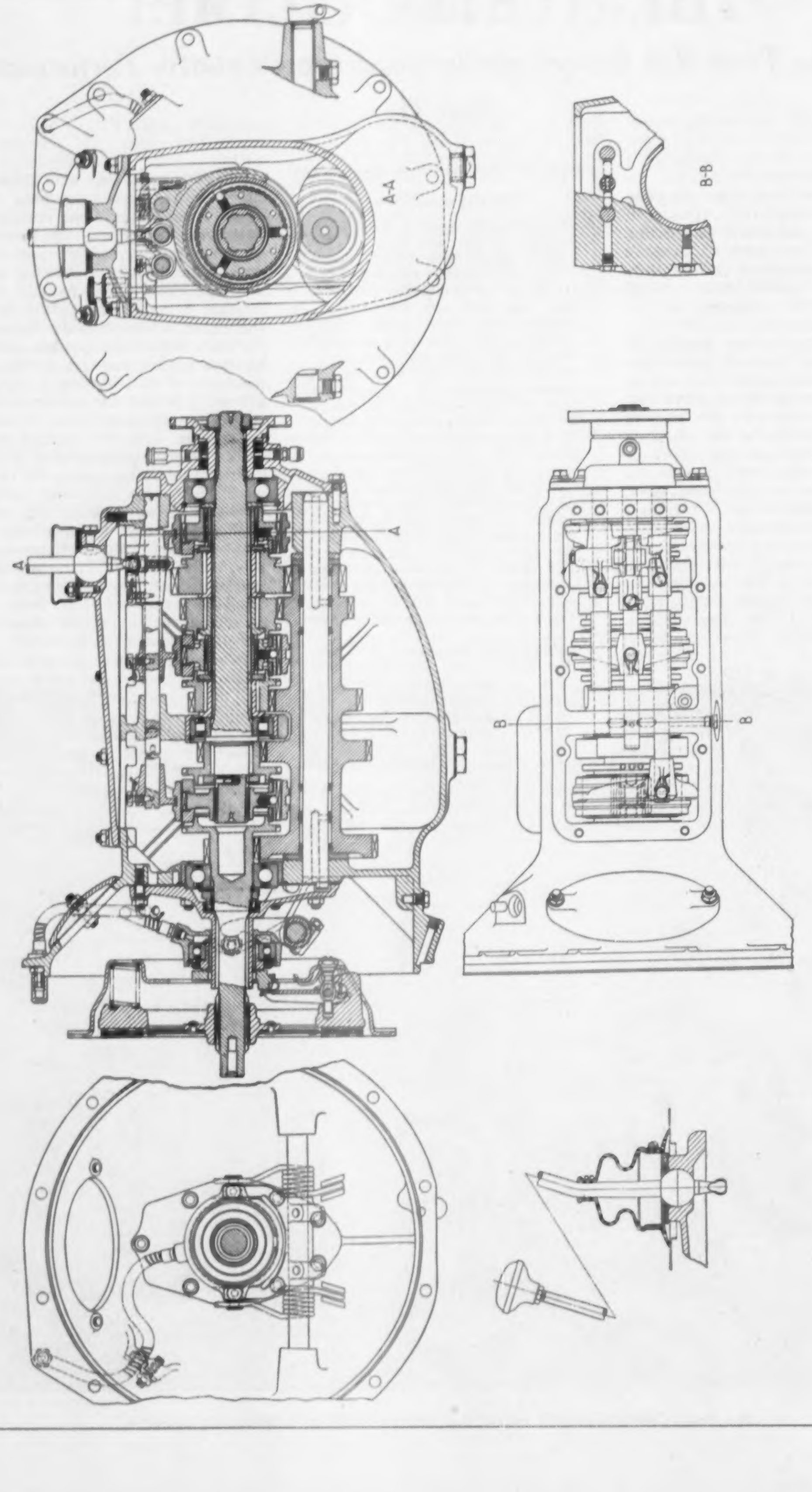
TRANSMISSION: Borg & Beck 10 in diameter single dry plate clutch. Overall ratios: fifth 4·98:1, fourth 7·62:1, third 11·84:1, second 17·71:1, first 27·25:1, reverse 26·53:1. Hypoid bevel rear axle with swinging half shafts. Final drive ratio: 1·02:1 through transfer gear train, and 7:34 through crown wheel and pinion.

SUSPENSION AND STEERING: Double-transverse-wishbone-link independent front and rear, with torsion bar springs and rubber assister springs. Armstrong A.T.7 shock absorbers with 1 in diameter bores. Austin rack-and-pinion steering unit giving $2\frac{1}{4}$ turns of the wheel from lock to lock and a turning circle of 35 ft.

BRAKES AND TYRES: Girling hydraulic brakes, two leading shoe at the front and one leading and one trailing shoe at the rear. Drum diameter 10 in and shoe width $1\frac{3}{4}$ in. 6.50—16 tyres maintained at 27 lb/in^2 pressure front and rear. Wheels, 5.00 F—16, with $1\frac{9}{16}$ in offset one-piece well-base type rims.

DIMENSIONS: Wheelbase 7 ft. Track, front and rear, 4 ft. Ground clearance 10 in. Overall length 12 ft $0\frac{1}{4}$ in, width 5 ft $1\frac{1}{2}$ in., height 5 ft 11 in. Frontal area 3,355 in^2. Dry weight 3,122 lb.

The Austin Champ is a sturdy vehicle, designed specifically for cross-country operation



ARRANGEMENT OF THE GEARBOX
All five forward speeds are fitted with synchro units

overall diameter of the toothed clutch is about $3\frac{3}{4}$ in while the mean diameter of the cone is approximately $2\frac{7}{8}$ in.

Two bearings carry the En 36-V18A mainshaft: one is of the ball and the other of the roller type. Axial location is effected at the rear by the $1\frac{3}{16}$ in inside diameter ball bearing. The outer race of this bearing is in a $3\frac{9}{16}$ in diameter housing, machined in the rear cover, and is retained between a shoulder in this housing and a snap ring. A nut on the $\frac{7}{8}$ in diameter rear end of the shaft retains the assembly comprising the companion flange for the universal joint, the inner race of the ball bearing, the first, second and third speed gears and the inner race of the roller bearing near the front end of the shaft.

This race bears against a collar on the shaft, and its inside diameter is $1\frac{9}{16}$ in. The outer race is $3\frac{3}{16}$ in outside diameter and is in a cupped housing carried in an aperture in a transverse web in the gearbox. It is retained between an inwardly turned lip at one end of the housing and a snap ring in a groove at the other. The lower portion of the housing is machined to clear the adjacent layshaft gears, and positive location is effected by a dowel-ended set-screw that registers in a hole in the housing. To allow for expansion and contraction of the shaft, the assembly comprising the inner race, rollers and cage is allowed a certain amount of float relative to the outer race.

The fourth speed gear and the synchronizer unit for fourth and fifth speeds are immediately forward of the roller bearing. On the $1\frac{3}{16}$ in diameter splined front end of the main shaft is the En 25V centre component of the synchronizer. The D.T.D.174A baulk rings are carried in annular recesses in the front and rear faces of the centre component. The included angle of their conical faces is 25 deg. Each of these two rings has three radial lugs spaced equally round its periphery. These lugs project through slots in the rim of the centre component and register in a groove formed by removing three splines, 120 deg apart, from the En 36V sleeve component. The inner ends of these lugs are machined at an included angle of 120 deg and the ends of the adjacent splines in the sleeve component are machined in a similar manner to form a lead-in for the lugs during the synchronizing motion. Three spring-loaded ball-type detents in the centre component register in a groove in the sleeve in the usual manner.

Synchronization is effected as follows. The initial movement of the sleeve causes the centre component and the baulk rings to slide axially until the cone is engaged. If the two shafts are out of synchronization, the baulk ring rotates through a small angle relative to the sleeve until the clearance between the lugs and adjacent splines between which they project is taken up. This sets the splines in the sleeve out of phase relative to the toothed clutch so that the two cannot engage. When synchronization is complete, the lugs on the cup rings ride in the 120 deg lead-in on the splines until they are centralized. It is then possible for the sleeve to be moved further until the toothed clutch is engaged. The leading edges of all the teeth of the clutch are chamfered to facilitate this final motion into engagement.

Behind the synchro-unit, the integral fourth speed gear, toothed clutch and synchro-cone are carried on 29·8 mm long needle roller bearings on a $1\frac{3}{16}$ in diameter portion of the shaft. They are retained by a 0·281-0·282 in thick internally splined washer locked by a spring-loaded plunger in a hole drilled radially in the shaft. This washer is assembled in the following manner over the splines that carry the centre component of the synchro unit. The plunger is depressed so that the washer can be moved over the plunger until it comes against the gear. Then the washer is rotated in the groove round the shaft until the plunger springs out to register between the splines. This locks the washer in the groove, with its splines out of normal phase with those round the shaft.

The third, second and first speed gears, as well as their synchronizers, are carried on needle roller bearings, about $1\frac{7}{8}$ in inside diameter × $2\frac{1}{8}$ in outside diameter, round En 36V sleeves splined on to the $1\frac{13}{32}$ in diameter portion of the shaft. This sleeve arrangement is employed to simplify the machining of the shaft. All the sleeves, except those that carry the synchro-units, have flanges at their forward ends to act as spacers between the components. At the front ends of each of the two that carry the synchro-units, a 0·206-0·208 in thick, En 32B washer performs this function: this arrangement has been adopted, of course, because it would have been impossible to machine the splines on the sleeves for the synchronizers without cutting through the flanges.

Both the synchro-units are similar to that already described for fourth and top speeds, except, of course, that a single, instead of double, unit is used for first speed. An unusual feature of the gearbox is that all the toothed clutches and synchro-cones are the same diameter and, therefore, are directly interchangeable. The surfaces of the baulk rings have a thread machined in them and are radially slotted to break down the oil film during engagement.

Another unusual feature of this gearbox is that two lip type oil seals are fitted in the rear cover to bear on the boss of the companion flange for the universal joint. These seals are arranged back-to-back, so that one prevents oil from leaking out of the unit and the other prevents foreign matter from entering. If the outer seal were to run dry, it would deteriorate rapidly; to avoid this, a lubrication nipple is fitted so that oil can be injected into the space between the two seals. A relief valve is also fitted to prevent the seals being damaged by pressure due to over-lubrication. The end of the seal housing is enshrouded by a sleeve pressed on to the companion flange for the universal joint.

In front of the seals, the outer race of the rear ball bearing, on which the whole of the mainshaft assembly is carried, is housed in the D.T.D.133C die-cast aluminium alloy rear cover. It is retained in its housing by a snap ring. The cover is spigoted into an aperture in the end of the gearbox. This aperture is of such a size that the whole of the assembly can be withdrawn through it.

A dead layshaft of En 36T is employed, and all the layshaft gears are integral, in a cluster on the sleeve. The shaft is 1 in nominal diameter. It is carried in the end walls of the box and is located between the front and rear covers. At the front, the end of the shaft is shouldered and registers against a projection in the cover to prevent the shaft from rotating.

Four needle roller bearings, two at each end, carry the layshaft cluster. Each bearing is 23·8 mm long × 25 mm inside diameter × 31 mm outside diameter, and each pair is retained by snap rings in grooves in the bore of the cluster. A similar snap ring arrangement separates the two bearings in each pair. Three thrust washers are interposed between the ends of the cluster and the casing, one at the front and two at the rear. All are 0·216-0·218 in thick and are of En 18C. Two washers are necessary at the rear, to take up the clearance that is required for the withdrawal of the layshaft

GEARBOX DATA

All gears are of En 36V and have a helix angle of 30 deg

	Pitch-circle diameter in	Diametral pitch	Projected length of teeth in
Primary and mainshaft gears:			
primary	1·75	8/10	$1\frac{1}{8}$
4th	2·375	8/10	$1\frac{1}{8}$
3rd	3·0	8/10	$\frac{15}{16}$
2nd	3·625	8/10	$\frac{15}{16}$
1st	4·25	8	$\frac{31}{32}$
Layshaft cluster:			
layshaft, or constant mesh, gear	4·5	8/10	$1\frac{1}{16}$
4th	4·0	8/10	$\frac{15}{16}$
3rd	3·25	8/10	1
2nd	2·625	8/10	1
1st	2·0	8	1

cluster. Details of the gears are given in the accompanying table.

The front ends of the En 8D, range R, selector rods are carried in holes in the transverse web that houses the front bearing of the mainshaft; the other ends are supported in the rear wall of the box. All three rods are 0·5605-0·5615 in diameter. Spring-loaded plunger type gear locks are assembled in holes in the rear wall. The interlock is in the transverse web. It is a more or less conventional arrangement comprising three plungers in line: one is of smaller diameter than the others and is in a diametral hole in the centre rod; the other two are on each side of it, in holes in the housing. Grooves are machined in the rods in such a position that when the gears are in neutral, all are in line with the plungers. If the centre rod is moved to select a gear, the plungers on each side are forced outwards to register in grooves in the other two and thus lock them. On the other hand, if one of the other rods is selected all three plungers are moved away from it to lock the other two rods. Thus, two gears cannot be selected simultaneously.

All the selector and striker forks are located by conical-ended set-screws assembled radially into their bosses to register in holes in the rods. For the selector forks, En 3A is employed, and the striker forks are also En 3A. The selector fork for top and fourth speeds is on the front end of the left-hand rod and is overhung forward of the transverse web that supports the rod. All the others are assembled on to the rods between their end supports.

A D.T.D.133C die-cast aluminium cover is bolted on to the top of the box. At its rear end there is a turret in which the En 32 striker lever is housed. This lever projects vertically downwards, in the normal way, between arms of the striker forks on the rods. First speed is an emergency low gear and, to prevent its being engaged accidentally, a spring-loaded plunger is housed in the fork. To make the action of this plunger even more positive, a groove is machined round its outer end, and a second spring-loaded plunger, also housed in the fork, but perpendicular to the first, registers in this groove. Thus, before the first plunger can be moved, the second has to be forced out of the groove. With this spring-loaded plunger arrangement the gear shift is rather more positive and simple to operate than when the lever has to be lifted over a shelf. This is a marked advantage with a gear that is likely to be required only in circumstances in which it is most important that the selection can be effected easily and without hesitation

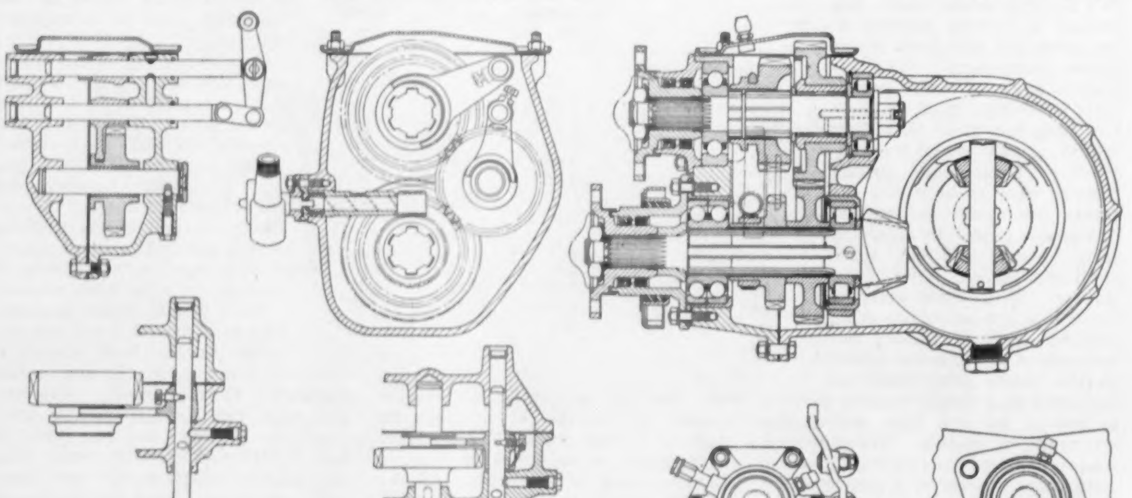
The lubrication arrangements in the gearbox are fairly conventional. It is recommended that S.A.E.30 grade oil be employed. The capacity of the box is 8 pints, and the static oil level is such that the sleeve of the layshaft cluster is just covered. In other words, the gears themselves are not entirely submerged. All the mainshaft bearings are lubricated by splash. Oil splashed through the front and rear bearings runs down through ducts in the inner faces of the end covers and into axial holes drilled from the ends of the layshaft. A radial hole, drilled into each of these axial holes from below, feeds the oil by gravity into the space between the two bearings at each end. Circular grooves are machined eccentrically in the faces of the thrust washers between the layshaft cluster and the end walls of the gearbox. Their function is to ensure that lubricant is spread over the whole of the thrust surfaces.

Rear axle

A Hardy Spicer propeller shaft, 2¼ in diameter × 36 in long, with needle roller universal joints, transmits the drive from the gearbox to the back axle. As has already been stated, the transfer and reverse gears are incorporated in the rear axle. Therefore, the nose-piece of the axle houses the idler spindle as well as the two shafts that carry the transfer gears. The drive from the gearbox is taken to the upper shaft, and the output to the front axle is connected to the lower shaft, on the rear end of which is forged the spiral bevel pinion.

The drive is transmitted from the upper to the lower shaft by a pair of constant-mesh gears. One of these gears is splined on to the lower shaft while the other is free to rotate on a needle roller bearing round the input shaft, to which it can be connected or disconnected by a toothed clutch. The centre component of this clutch, which also serves as the reverse gear, is splined on to the shaft in front of the floating gear. For the selection of the reverse speed the idler gear is slid into mesh with the teeth round the periphery of the centre component of the clutch and a second gear splined on the lower shaft. The controls for the engagement of forward and reverse drives are interconnected so that it is impossible to select both simultaneously.

Both the axle casing and the nose-piece, which is a separate bolted-on extension, are of D.T.D.133C. The two components are held together by seven $\frac{5}{16}$ in diameter clearance bolts and two $\frac{3}{8}$ in diameter fitted bolts. A Tecalemit breather, of the type already mentioned in connection with the gearbox, is fitted. The oil capacity of the unit is 4½ pints and S.A.E.90 grade is recommended. In the nose-piece, the input



The reverse gear is housed in the transfer box in the rear axle unit

Reverse and transfer gear assembly

shaft is carried on two bearings. A Ransome and Marles MDJT 40 single row ball type unit is used for the front bearing, while the rear one is an MR3 40 roller bearing. The toothed clutch and the constant-mesh gear are between the two bearings.

The shaft is of En 36V. It is approximately $1\frac{1}{4}$ in diameter where it carries the constant-mesh gear. At the front end, the shaft is 1·120-1·125 in diameter, and the splines for the companion flange for the universal joint are $\frac{1}{16}$ in deep. The portion of the shaft that carries the En 36Y sliding member of the toothed clutch is 1·362-1·636 in diameter and has splines machined in it to a depth of $\frac{1}{8}$ in. A smaller bearing is used at the rear partly because rollers have a higher load-carrying capacity than balls and partly because it is not loaded so heavily by the propeller shaft as is the front one. The outer race of the rear bearing is retained by a circular-section, snap ring in a groove in its housing in the casing.

At the front end of the shaft, the bearing is retained by an En 2A cover spigoted into its housing. This cover carries two oil seals assembled back-to-back, with a space between them. They bear on the boss of the companion flange for the universal joint. A nipple is screwed into the casing so that lubricant can be supplied between the seals, otherwise the outer one would run dry. A relief valve is also fitted to prevent damage to the seals as a result of over lubrication; it is also designed to prevent the entry of water when the vehicle is wading. The housing for the seals is enshrouded by a sleeve, which is flanged and bolted to the companion flange for the universal joint.

The companion flange is pulled against the inner race of the front bearing by a nut on the $\frac{7}{8}$ in diameter, threaded end of the shaft. Axial location of the assembly is effected at the front bearing, the inner race of which is clamped between the companion flange and a shoulder on the shaft. A limited amount of axial float is permitted between the outer race of the rear bearing and the assembly comprising the rollers, their cage and inner race. The inner race of the rear bearing is pulled on to the shaft by a split-pinned nut of the same size as that at the front end of the shaft. It bears against an En 32 distance washer located against a shoulder round the shaft.

The En 36V, constant mesh gear, which has a diametral pitch of 8/10, is pressed on to the flanged outer race of the needle roller bearing that carries it on the shaft. This arrangement has been adopted in preference to using the bore in the gear as the outer race, because the latter construction would have necessitated restricting the length of the bearing to give clearance for hobbing the internal teeth of the toothed clutch. There are two rows of needle rollers in this bearing. Each is about $\frac{3}{8}$ in long. An En 32 thrust washer is interposed between the flange at the front of this bearing and a shoulder on the shaft; there is another washer between the rear end and the inner race of the roller bearing that carries the shaft.

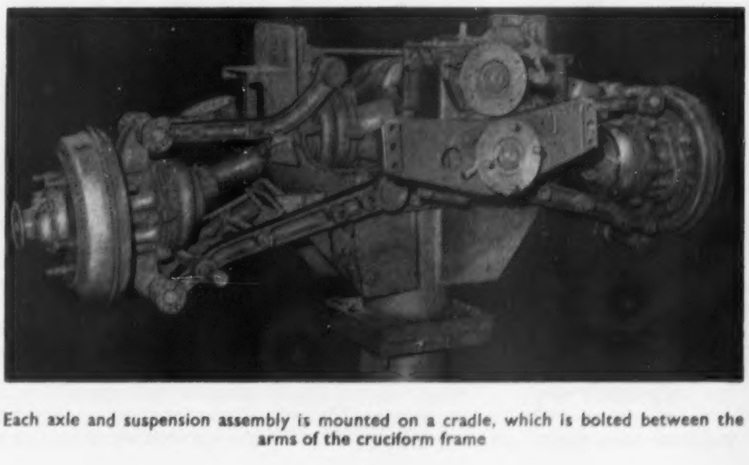
The shouldered portion of the shaft is splined to a depth of $\frac{1}{8}$ in and is 1·632-1·636 in diameter. The sliding gear that forms the centre component of the toothed clutch is of En 36Y. A cavity is formed at its rear end to clear the flange of the outer race of the bearing for the floating gear, and a channel is machined round its forward end to receive the En 3A selector fork.

A Clevite 25 bush, 0·868-0·874 in inside diameter by 1·049 in outside diameter, is pressed into the idler gear, which has a pitch circle diameter of 3·5 in. Axial location of the En 32B idler spindle is effected by a dowel-ended set screw, inserted from the outside of the case to register in a hole drilled radially in the spindle. The spindle is inserted from the front of the box and the hole through which it is passed is sealed by a cork ring. This ring is fitted round the projecting end of the spindle, and a circular plate retained by a countersunk set screw in an axial hole in the end of the spindle, is used to clamp it in position to effect the seal.

The front end of the lower shaft is 1·565-1·570 in diameter and is splined to a depth of $\frac{3}{32}$ in to receive the companion flange for the universal joint through which the drive is transmitted to the front axle. The hypoid bevel pinion is formed on the rear end. Loads on the lower shaft are the more severe, not only because it reacts the final drive torque but also because the propeller shaft to the front axle is longer than that between the gearbox and the back axle. Therefore it is carried in larger bearings than the upper one.

At the front, an HJ30, double-row ball bearing is employed; the rear one is an MRJ 30 roller bearing. The outer races of both bearings are in separate En 2A housings. This arrangement has been adopted so that the shaft assembly can be withdrawn complete from the casing. The shaft is 1·565-1·570 in diameter over most of its length and has six splines, $\frac{3}{32}$ in deep, machined in it. Assembled on to it in the following order are: a distance washer, which bears against the pinion; the rear bearing; the forward drive gear; reverse drive gear, on a sleeve extension of which is mounted the speedometer gear; the front bearing; a distance piece; and the companion flange for the universal joint. This assembly is retained by a nut on the $\frac{7}{8}$ in diameter threaded front end of the shaft.

Axial location of this assembly is, of course, effected by the two-row ball bearing at the front. This bearing has a single-piece inner race and a two-piece outer race. A flange round the housing that contains the outer race is clamped between the front En 2A cover and the front wall of the box. This front cover houses two oil seals arranged in a similar manner to those in front of the ball bearing that carries the upper shaft. The sleeve extension of the housing for these seals is longer than that round the upper shaft, because it is supported in a rubber ring housed in a sleeve in the front wall of the cradle that carries the whole of the final drive casing. To reduce the manufacturing costs all the companion flanges for the universal joints are identical. It is for this reason that an En 8 distance piece is fitted between the boss of the lower flange and the inner race of the ball bearing. The rear bearing housing is located in the aperture in the front wall of the final drive casing by a dowel-ended set-bolt screwed in radially from the outside of the casing to register in a hole in the housing. The outer race is retained in the housing



Each axle and suspension assembly is mounted on a cradle, which is bolted between the arms of the cruciform frame

by a rectangular section snap ring.

En 325 is used for the reverse gear, which has a diametral pitch of 8 and a pitch circle diameter of 3·625 in. Mounted on its boss is the speedometer gear, which is of En 34. Behind this assembly is the En 36V forward speed gear, which has a diametral pitch of 8/10. All the gears in this box, except those in the reverse train, have a 30 deg helix angle. The reverse gears are of the straight spur type, since the idler meshes with the centre component of the toothed clutch.

In designing a mechanism for the selection of forward and reverse drives, the problem was to develop a simple means of changing from one direction to the other, while at the same time making it impossible for both gears to be engaged simultaneously. Two $\frac{7}{16}$ in diameter, En 8D selector rods are installed horizontally, one above the other, in the nose-piece extension of the axle casing, and a conventional, plunger type interlock is carried between them. The upper rod is for the selection of forward drive, the lower for reverse, and the interlock is so arranged that when either is engaged the plunger is forced into a groove in the disengaged rod, which it thus locks. Cupped steel plugs seal the rear ends of the holes in which the tail ends of the rods are carried, and rubber rings in annular grooves in the front pair of holes seal the forward ends.

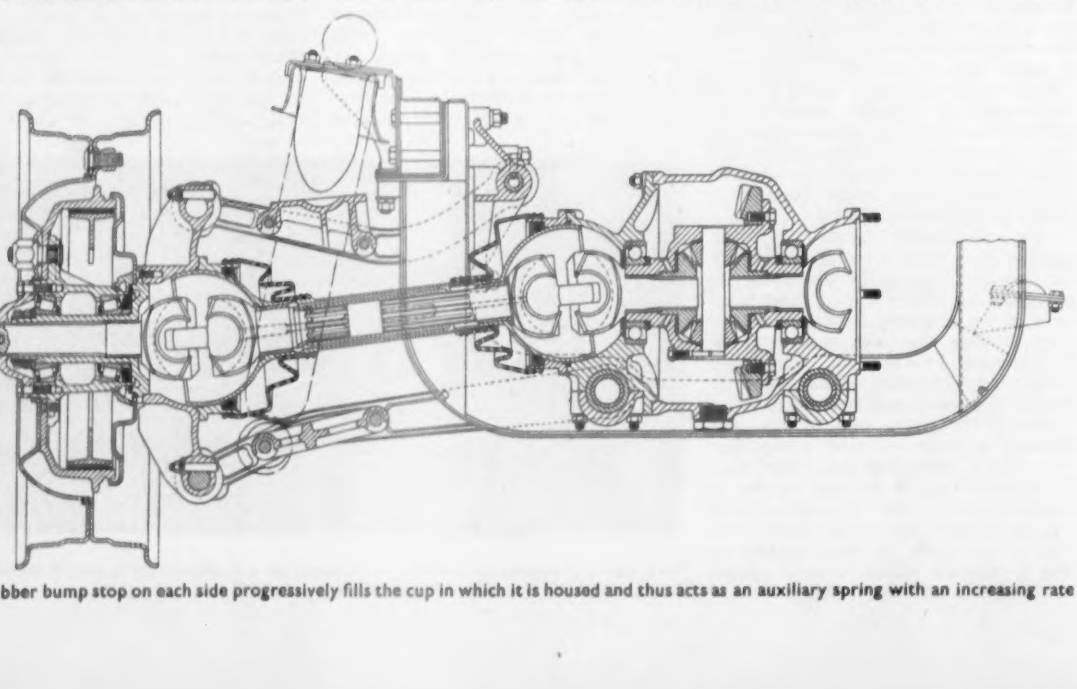
The En 3A lever, by means of which the rods are actuated, is pivoted on the upper of the two rods, that is, the rod used to engage or disengage the forward drive. The other end of this lever is connected by a shackle to the lower rod, for the selection of reverse speed. A control rod is pinned to the upper end of the lever. As can be seen from the illustration, if the control rod pushes the upper end of the lever to the rear the tendency is for the upper rod to move in the same direction and the lower one to be pulled forwards. If reverse gear is in engagement, this forward motion of the lower rod disengages it; then, when this rod has moved the full extent of its travel, the plunger of the interlock is in line with a groove machined in it, so the upper rod can then be moved rearwards to engage forward speed. The sequence of events is exactly the opposite when the other end of the control lever is pulled forwards to change from forward to reverse speeds.

The En 3A selector forks are secured to the rods by means of conical-ended set screws assembled radially into tapped holes in their bosses to register in holes in the rods. Spring-loaded-plunger type locks register in grooves in the rods to ensure that the gears do not slip out of engagement after they have been selected. In addition, to prevent selection of reverse while the drive is being transmitted in the forward direction, the thrust faces towards the rear ends of the splines that carry the sliding component of the toothed clutch are relieved by about 0·004 in, in such a manner that the sliding component has to ride over the shoulders at the junction between the relieved and unrelieved portions before forward drive can be disengaged. To facilitate production, a groove is machined round the shaft to separate these two portions.

Lubrication of the gears and bearings in the final drive casing and nose-piece is entire by splash. The capacity of the unit is $4\frac{1}{2}$ pints, and S.A.E.90 Grade lubricant is specified. Oil thrown forward off the top of the crown wheel is splashed into the two roller bearings that carry the rear ends of the shafts in the nose-piece. Some also passes into an axial hole drilled in the end of the upper shaft, whence it drops vertically through a radial hole into the space between the two rows of needle rollers that carry the constant-mesh gear. In addition, a duct is cored in the wall between the final drive casing and its nose-piece at a point just above the level of the meshing-point of the constant-mesh pair of gears, and oil splashes from the crown wheel through this hole to supplement the splash from the gears in the nose-piece. This is a good feature since this pair of gears is loaded almost continuously, whereas the others in the nose-piece, namely those of the reverse train, are only in use for relatively short periods. A duct is also cored in the base of the dividing wall to allow the oil to flow freely between the two compartments to maintain the same level in both. The two ball bearings at the front of the nose-piece are lubricated by splash from the gears adjacent to them.

The final drive casing is mounted on a rubber bush round the tubular housings for the pivot pins of the lower wishbones. Two semi-circular seatings, machined one on each side in the lower face of the casing, rest on the rubber bushes; semi-circular caps are bolted up from below to complete the assembly. The ends of the housings for the pivot pins are welded in holes in the front and rear walls of the cradle that carries the final drive casing and suspension units. This cradle is bolted to the arms of the cruciform frame. As has already been mentioned, additional support, to take drive- and brake-torque reaction, is afforded by a rubber mounting round the nose-piece that houses the seals at the front end of the output shaft from the transfer box to the front axle.

Both the final drive casing and the nose-piece are of D.T.D.133C. The final drive casing is of the pot type, with a cover spigoted on to its left-hand side. This cover is also of D.T.D.133C. It houses the bearing that supports one side of the differential cage, and also forms part of the spherical housing for the universal joint at the inner end of



The rubber bump stop on each side progressively fills the cup in which it is housed and thus acts as an auxiliary spring with an increasing rate

the swinging half shaft. The bearing housing and spherical half-housing on the other side of the unit are formed in the casing itself.

There are seven teeth on the crown wheel pinion, the axis of which is $1\frac{1}{4}$ in below that of the crown wheel and in line with the plane that contains the axis of the differential pinion spindle. The axial position of the crown wheel pinion is controlled by shims between the pinion and the distance washer that separates it from the rear bearing. Rigidity of the support for the pinion is ensured by the fact that the bearings that carry it are spaced with their centres $4\frac{3}{4}$ in apart.

The 8·267 in diameter En 35 crown wheel has 34 teeth. This wheel is assembled over the one-piece, B.S.310 grade 2, cast iron differential cage and spigoted on to a flange round the cage. It is secured by twelve, wire-locked set screws, $\frac{3}{8}$ in diameter. The $\frac{7}{8}$ in diameter, En 32B differential pinion spindle is pressed into the cage. It is located by a dowel-ended set screw assembled into a tapped hole in the cage to register in a diametral hole drilled through the spindle.

Two En 398 differential pinions are employed. They mesh with the 3·520 in diameter, En 36 differential gears. The length of tooth engagement is approximately $\frac{7}{8}$ in, and the backlash between these gears and pinions is 0·008-0·010 in. A CPIA phosphor bronze spherical washer, $\frac{7}{8}$ in inside diameter × 2 in outside diameter, is interposed between the outer face of each differential pinion and its seating in the cage. Flat thrust washers of the same material but $3\frac{1}{4}$ in outside diameter × $1\frac{3}{4}$ in inside diameter, are interposed between the outer faces of the differential gears and the cage. These washers seat in recesses in the cage, which locate them radially. The bosses of the differential gears bear for a length of about $\frac{7}{16}$ in in the $1\frac{3}{8}$ in diameter counterbore in the cage, and $\frac{3}{16}$ in diameter holes are drilled from the outside of the cage to break through this bearing surface and supply it with oil.

Each side of the cage is machined to receive the $2\frac{1}{2}$ in inside diameter, roller bearings that support it in the casing. Distance washers are interposed between the inner races of these bearings and shoulders on the cage. They are incorporated for adjusting the mesh of the crown wheel and pinion. The nip of the bearings is regulated by shims between the cover and the final drive casing. Tracta universal joints are employed at both the inner and the outer ends of the swinging half shafts. They are of En 8. At the inner end of each half-shaft, the stub-shaft on the driving half of the universal joint is machined to about $1\frac{1}{4}$ in diameter and splined into the differential gear. Immediately outboard of the gear, the stub-shaft is supported by a Vandervell Clevite bush, 1 in long, pressed into the differential cage.

The swinging half-shaft and sliding joint are enclosed in a tube. On the outer end of the tube, a hemispherical cup is formed. Together with another hemispherical cup on the wheel assembly, it forms a spherical housing to enclose the universal joint. A similar arrangement is used to protect the universal joint at the inner end; but, in this case, the outboard hemispherical cup is a separate component, of En 8, pressed into the end of the tube, and the inboard cup is formed in the final drive casing. At both ends, the cup on the tube—that is the inner one—is retained in the outer one by a ring-shaped casting of DTD 424. These retainer rings are spigoted into the outer cups to locate them accurately to ensure continuity of the spherical surface. Lip type oil seals are fitted in them and bear on the outer surface of the cups on the tube. Rubber gaiters are fitted over the seal housings and are clamped to the tubes.

The sliding joints are simply En 16T sleeves into which the stub-shafts of the universal joints are splined. Axial and radial holes are drilled in the stub-shafts to equalize the pressures throughout the housing and thus prevent the pumping action of the sliding joint from forcing oil into the rubber gaiters. At the outer end of each sleeve is a counterbore to receive a collar round the stub-shaft. After the shaft has been assembled into the sleeve until the collar bottoms in the counterbore, the end of the sleeve is rolled into a groove round the shaft immediately outboard of the collar. This locks the sleeve on the outer stub-shaft. Both stub-shafts are $1\frac{1}{4}$ in nominal diameter, and the sleeve is about $1\frac{7}{16}$ in outside diameter. The shafts are carried in $1\frac{1}{2}$ in long, Oilite FBS98 bushes, adjacent to the universal joints, which are flanged and fitted in the necks of the hemispherical cups. The inside diameter of the bush at the inboard end is approximately $1\frac{1}{4}$ in, but the bush at the outboard end has been made $1\frac{3}{8}$ in diameter so that the sleeve of the sliding joint can be passed through it on assembly.

At the outboard end of the swinging half shaft assembly, the outer half of the spherical housing for the universal joint is formed in the En 25 stub-axle carrier. This carrier is spigoted on to a flange at the inner end of the En 49B stub-axle, and the axle is of tubular form so that the stub-shaft of the outer universal joint can be passed through it on assembly and splined into an En 8 driving flange. This driving flange is retained by a dished plate secured by a conical seating set screw assembled into a tapped hole in the end of the shaft. It is spigoted and bolted to the B.S.310 grade 2 cast iron, wheel hub.

The hub is carried on two, Timken 359 S/354 A taper roller bearings spaced with their centres about $2\frac{1}{2}$ in apart. These bearings are assembled into it from each end, and the whole assembly is secured to the stub-axle by a ring nut. A second ring nut on the $1\frac{1}{4}$ in diameter end of the stub-axle is tightened against the first, to lock it; as a further precaution, a tab washer is fitted between the two. A lip type oil seal is housed in the inner end of the hub and bears on a distance ring between the inner race of the inner bearing and a shoulder on the stub-axle. The main reason for the employment of a distance ring, in conjunction with the seal, is to enable a large radius fillet to be incorporated, but it also has two other advantages. One is that, when worn, it can be readily replaced; the second is that the fine grinding necessary in the area on which the seal bears is restricted to the ring.

A B.S. 1452 grade 17 brake drum is carried in the usual manner on a flange round the hub. It is spigoted-on and secured, together with the wheel, by five $\frac{3}{8}$ in diameter studs and conical seating nuts. The brake back-plate is spigoted on to a flange round the inner end of the stub-axle, to which it is bolted, together with a die-cast ring. This ring acts as a trap to prevent any lubricant that might escape past the seal in the hub from getting into the brake drums. Drainage holes are drilled from the groove round the inner periphery of the ring to the outside of the brake unit. A chrome-leather washer is interposed between the die-cast ring and the end of the spigot. It bears on the outer periphery of the seal housing to keep out abrasive matter.

Rear suspension

Torsion bar type springs are employed in the independent rear suspension and damping is by AT7 shock absorbers. The rubber bump stops are, in fact, auxiliary springs. They are housed in pressed steel cups and, as they deflect, they gradually fill the cups so that the spring rate increases with the deflection.

The unsprung weight is 151 lb per side. The height of the roll centre is $7\frac{3}{4}$ in above ground. In the unladen condition, the rear end rate of the vehicle is 426 lb/in. This gives a periodicity of 105 c/min. The deflection to the laden position is 3·3 in, and the deflection to full-bump is $7\frac{5}{8}$ in.

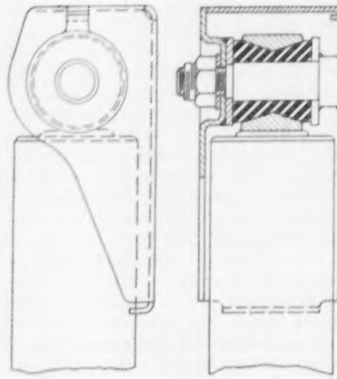
A double-transverse-wishbone-link type of layout has been adopted. Both the upper and the lower wishbones are two-piece forgings. On the lower wishbones, the arms are of H-section. At their outer ends the section is $\frac{7}{8}$ in wide over the flanges by $1\frac{5}{16}$ in deep, and at their inner ends it is $1\frac{5}{16}$ in wide over the flanges and 2 in deep. The web thickness of the rear arm is $\frac{5}{16}$ in, while that of the front arm is $\frac{7}{16}$ in. The front arm is thicker than the rear one, because the torsion bar spring is attached to it. A distance piece is interposed between the two arms near their outer ends, where they are clamped together by two $\frac{1}{2}$ in diameter bolts. The distance between the outer ends of the arms is controlled by shims on each side of the distance piece.

The geometry of the system is as follows. As measured horizontally, the distance between the pivot centres of the lower wishbone arms and the centreline of the chassis is $4\frac{1}{8}$ in. The upper wishbone pivot axes are $7\frac{3}{4}$ in

from the centreline of the chassis, and the distance as measured vertically between the upper and lower pivots is 8 in. At the outer ends of the wishbone arms, the distance of the axis of the lower pin from the centreline of the chassis is $19\frac{7}{8}$ in, while the distance from the axis of the upper pin to the centreline is $20\frac{1}{16}$ in. Between the two, the vertical spacing is $10\frac{9}{16}$ in. At the inboard end of the bottom wishbone on each side, the distance between centres of the pivot bearings is $10\frac{3}{8}$ in, and between the bearings at the outer end it is $3\frac{13}{16}$ in. On the upper link, these distances are 9 in and $3\frac{15}{16}$ in respectively.

At the inboard end of the lower wishbone, the En 8R pivot pin is 0·8735-0·874 in outside diameter; at the outboard end, the pin is 0·874-0·87475 in diameter and is of En 361. The pivot pins of the lower wishbones are in tubular housings, the ends of which are welded into holes in the cradle that carries the final drive and suspension units. As has already been mentioned, the final drive casing is mounted on these tubular housings. Each housing is fabricated from three pieces: one is the central tube, to which the other two—the end bosses in which the pin is carried—are butt-welded. The pin is located by a conical-ended set-screw, which is assembled into a tapped, radial hole in the rearmost boss to register in a hole in the pin.

Oilite Superload flanged bushes are employed for the pivot bearings. They are assembled into the inner ends of the bosses on the wishbone arms. The outer end of each of these bosses is sealed by a circular plate, which is peened in. A thrust washer is interposed between the flanged inner ends of each bush and the housing that encloses the pins. It is prevented from rotating by a small peg passed through it and driven into a hole in the end of the housing. To prevent abrasive matter entering and oil escaping from the housing, a spring-loaded, rubber sealing ring is fitted round the flange and the end of the housing. This ring is enshrouded by a sleeve pressed on to the bearing boss on the end of the wishbone arm. The pin is drilled axially about $\frac{11}{32}$ in diameter to form an oil reservoir; the unit is filled with oil on assembly, and no further lubrication is required throughout the life of the vehicle.



Attachment of the upper end of the shock absorber

The internally splined boss that carries the end of the torsion bar is of BS.310 grade 2, malleable cast iron. It is spigoted into a counterbore in the forward end of the front wishbone arm pivot bearing housing. An arm on this boss is drilled for the two bolts that secure it to the wishbone arm.

At the outer end of the lower wishbone link, the 0·874-0·87475 in diameter, En 361 pin is carried in an eye at the bottom of the stub-axle carrier. It is located axially by a taper cotter that registers in a groove machined approximately mid-way between the ends of the pin. This eye and pin assembly is carried between the ends of the arms of the wishbone. Flanged, screw-type bushes are assembled on to the ends of the pin and screwed into the bosses in the ends of the wishbone arms. Each is secured by a bolt passed through one of the holes in its flange and screwed into the boss. The outer ends of both bushes are sealed by peened-in discs. A hole is pierced in the centre of the rear disc, and a lubrication nipple is screwed into it. A $\frac{1}{4}$ in diameter hole is drilled axially through the pin so that lubricant can pass from the rear end to the front. Rubber seals are fitted over the ends of the eye on the stub-axle carrier and are spring-loaded to bear on the inner ends of the bosses that carry the screwed bushes.

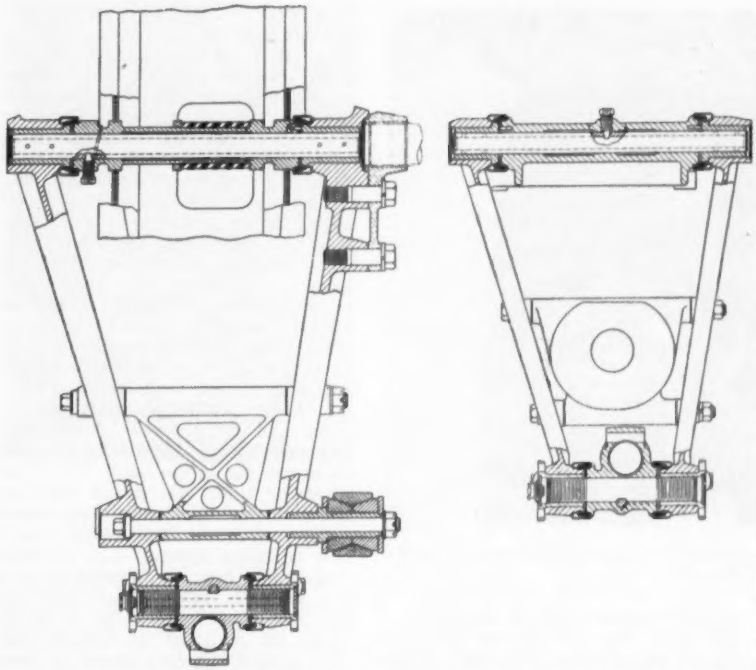
The rubber-bushed eye-type end fitting at the base of the telescopic shock absorber is carried on a sleeve round the $\frac{1}{2}$ in diameter bolt that clamps the two arms of the wish bone link against the distance piece which is interposed between them. This sleeve is spigoted into a counterbored boss on the front face of the forward wishbone link. The outside diameter of the sleeve is $\frac{7}{8}$ in and its overhang from the boss into which it is spigoted is 3 in. A split-pinned nut on the end of the bolt retains the rubber bush assembly. This assembly comprises two washers, between which the two-piece conical rubber bush is compressed. The rear washer is located by a collar round the sleeve and the other one is positively located by a shoulder on the sleeve. Thus, when the nut is tightened the pre-compression on the bush is limited.

The upper wishbone link is similar to, but smaller than, the lower one. It is also lighter because not only is it shorter but also it does not have to transmit the vertical loading to the torsion bar spring, as does the lower link. The minimum cross-sectional dimensions of each arm of the link are $1\frac{1}{8}$ in deep × $\frac{3}{8}$ in wide over the flanges and the web thickness is $\frac{5}{32}$ in.

At the outer end of the link the assembly is similar to that already described at the outer end of the lower link In fact, the bushes, seals, cotter and pivot pin are all interchangeable. The distance piece between the arms of the wishbone is of a slightly different shape since it forms the abutment for the bump stop rubber.

At the inner end of this link the pivot arrangement again is similar to that on the lower link, but, of course, there is no connection to the torsion bar. Also, the housing for the pin is a one-piece, BS.310 grade 2 malleable iron casting bolted to the inner face of the vertical member of the cradle that carries the final drive and suspension units. To secure this casting four $\frac{3}{8}$ in diameter bolts are passed through the cradle and the cruciform member of the underframe. Thus, they not only perform the function of anchoring the pivot pin housing to the cradle but they also attach the cradle to the cruciform. The pivot pin is 0·7485-0·749 in diameter, and it is drilled $\frac{7}{16}$ in diameter axially to form an oil reservoir. The reservoir

Rear suspension; showing the method of attachment of the torsion bars



Oil reservoirs are formed in the hollow pivot pins of the transverse links of the suspension units

is filled with oil on assembly and no further attention is required throughout the life of the vehicle. A conical ended set screw is used to locate the pin, but it is screwed in at the centre instead of near one end as in the pivot of the lower wishbone link. In both links two diametral holes at each end of the pin feed the lubricant into the bearing surface between the pin and the bush.

Front axle and suspension

The final drive casing of the front axle is similar to that of the rear. However, instead of a transfer box and reverse gear train, it houses in a separate casting, bolted to its nose-piece, the throw-out gear for the front wheel drive. The axle casing, like that at the rear, is of D.T.D.133C, but the throw-out casing is of die-cast D.T.D. 424 aluminium alloy. Inside the casing, the crown wheel, differential cage, and the differential pinions and gears are all identical with those at the rear; only the crown-wheel pinion is different.

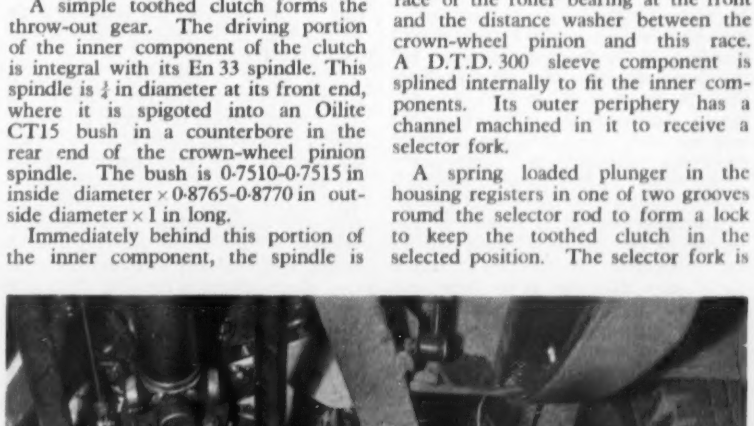
This pinion is integral with its $1\frac{9}{16}$ in diameter spindle. It is carried by a roller bearing adjacent to the pinion and by a two-row ball bearing at the rear. A tubular distance piece separates the two bearings, the centres of which are approximately $3\frac{5}{8}$ in apart. The outer race of the roller bearing is housed in the nose-piece casting, where it is retained by a snap ring. On the other hand, the two-piece outer race of the ball bearing is in a separate, flanged, cupped housing. This housing is bolted, together with a retainer ring that holds the outer race in position, to the rear face of a flange round the nose-piece of the main casing. Spigoted over it is the throw-out gear casing.

A simple toothed clutch forms the throw-out gear. The driving portion of the inner component of the clutch is integral with its En 33 spindle. This spindle is $\frac{3}{4}$ in diameter at its front end, where it is spigoted into an Oilite CT15 bush in a counterbore in the rear end of the crown-wheel pinion spindle. The bush is 0·7510-0·7515 in inside diameter × 0·8765-0·8770 in outside diameter × 1 in long.

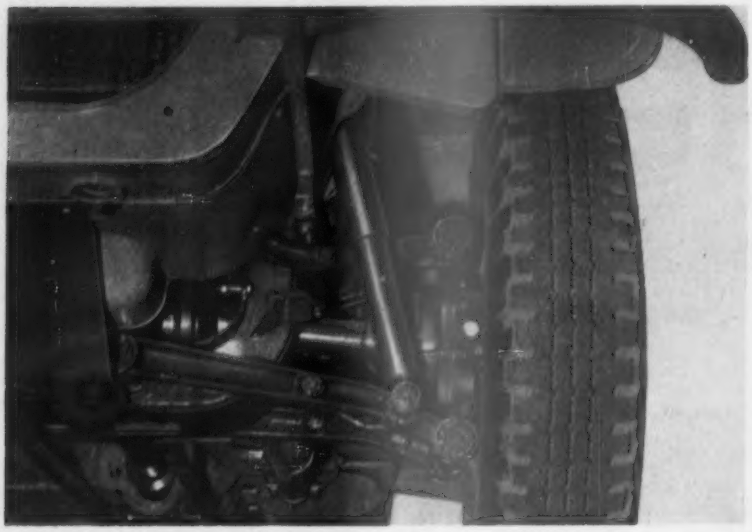
Immediately behind this portion of the inner component, the spindle is carried in a single row ball bearing. The inside diameter of this bearing is $1\frac{3}{16}$ in and the outside diameter $2\frac{7}{16}$ in. Its inner race is held in position by the companion flange of the universal joint, which is retained by a nut on the $\frac{7}{8}$ in diameter threaded rear end of the spindle. The outer race of the bearing is assembled into the casing from the front and retained by a snap ring. Immediately behind it are two oil seals of the lip type, housed back to back. A nipple is provided to feed lubricant into the space between the two seals, which bear on the boss of the companion flange. This is to prevent the outer one from running dry and failing prematurely. As at the rear, a relief valve, designed to prevent the entry of water, is also incorporated, to prevent excessive pressures from damaging the seals. A flanged sleeve is bolted to the companion flange and extends forwards to enshroud the seal housing. A rubber bushed support, similar to that used on the rear axle, steadies the nose-piece. The rubber bush surrounds the casing between the rear bearing and the oil seal assembly.

The En 33 driven component of the toothed clutch is splined on to the end of the crown-wheel pinion spindle and retained by a ring nut on the $1\frac{1}{8}$ in diameter threaded tail end. This nut pulls the driven component against the inner race of the two-row ball bearing and retains the whole assembly, including the distance piece, the inner race of the roller bearing at the front and the distance washer between the crown-wheel pinion and this race. A D.T.D. 300 sleeve component is splined internally to fit the inner components. Its outer periphery has a channel machined in it to receive a selector fork.

A spring loaded plunger in the housing registers in one of two grooves round the selector rod to form a lock to keep the toothed clutch in the selected position. The selector fork is



Front suspension; showing the steering layout



Under each arm of the upper wishbone link, a rubber rebound stop is fitted to a bracket on the cradle

floating on the rod except inasmuch as it is constrained by a compression spring to bear against a snap ring in a groove round the rod. This spring is located radially by an extension of the boss of the fork, and its rear end bears against a collar located against a shoulder on the rod. It is incorporated to facilitate engagement of front wheel drive when the vehicle is in motion. If, when the control is moved to select four wheel drive, the driving and driven members of the toothed clutch are not in alignment, the rod simply slides through the boss of the fork, and the spring is compressed. Then, when the two members move into alignment, to complete engagement the compression spring slides the fork forward until it is stopped by the snap ring round the rod.

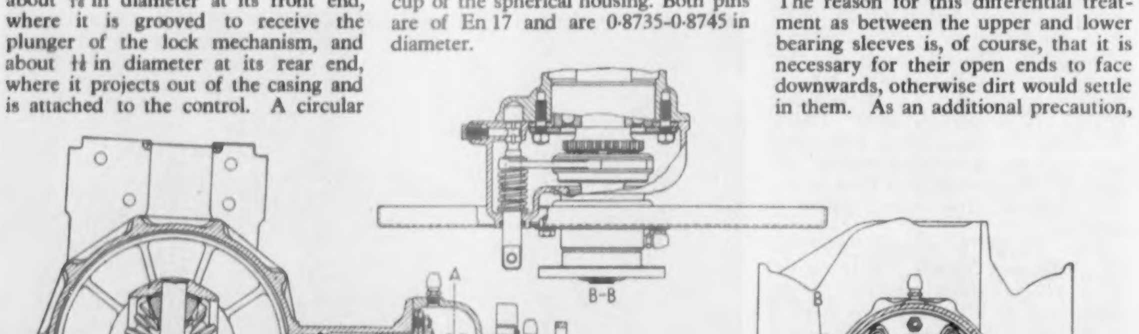
The selector rod is housed in two bosses in the die-cast casing for the throw-out gear It is of En 15 and is about $\frac{9}{16}$ in diameter at its front end, where it is grooved to receive the plunger of the lock mechanism, and about $\frac{11}{16}$ in diameter at its rear end, where it projects out of the casing and is attached to the control. A circular section rubber ring is housed in a groove in the bore of the boss that carries the rear end of the rod. There is adequate side clearance between the ring and the groove to allow for growth of the rubber under the influence of oil, but the ring is nipped radially by about 0·005 in to effect a seal against the ingress of water when the vehicle is wading. The boss that carries the front end of the rod and the lock is sealed by the final drive casing, which covers its forward end.

The drive to the front wheels is similar to that to the rear wheels. In fact the only major difference is that the swivel pins are formed on the spherical housing for the outer universal joints on each side. The lower component of each swivel pin is integral with the stub-axle carrier, while the upper component is formed on the ring that is bolted to this carrier to retain the inner cup of the spherical housing. Both pins are of En 17 and are 0·8735-0·8745 in diameter.

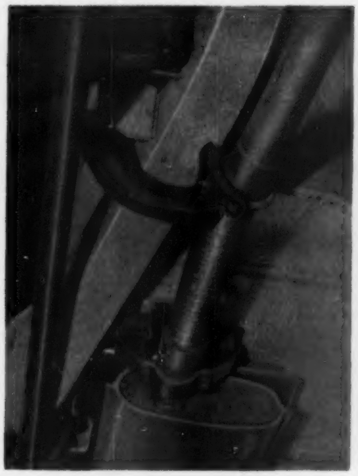
Screwed-sleeve type, swivel pin bearings of En 33 are employed. The upper one is $1\frac{19}{32}$ in long and the length of the lower one, $2\frac{1}{8}$ in. Their centres are about $10\frac{1}{4}$ in apart, as measured along the axis of the pins. They are flanged at one end and threaded externally; not internally, as is more usual. Each sleeve is assembled over the pin, with its flange at the end nearest to the spherical housing. Its other end is slotted to receive a tongued collar, which is passed over the end of the pin. There are two flats on the periphery of the pin and the hole in the collar is slotted to register on each side of these flats. Thus, the collar prevents rotation of the sleeve relative to the pin. The whole assembly is retained by a split-pinned nut on the end of the pin.

Screwed on to each sleeve is a phosphor bronze knuckle that pivots about the pin in the outer end of the wishbone link. To ensure that the knuckles are correctly positioned axially relative to their swivel pins, a hole is drilled through each to break tangentially into its threaded bore; a $\frac{3}{8}$ in diameter bolt is assembled into the hole and registers in a groove round the centre of the flanged bush. This groove is wide enough to allow for the thirty-two degrees of movement of the wheel on lock, but not to permit the bush to be assembled in the wrong position. The pivot pins at the outer ends of the wishbone links are secured to the knuckles by taper cotters that register in grooves in the peripheries of the pins.

Cupped steel caps are screwed over the outer ends of the knuckles to prevent abrasive matter from entering the swivel pin bearings or lubricant from escaping. Over the inner ends, spring loaded, rubber seals are fitted. They bear on the flanges of the bushes. Each seal is enshrouded by a sleeve, which at the lower bearing is pressed on to a shoulder round the inner end of the swivel pin, and at the upper bearing is pressed on to the knuckle. The reason for this differential treatment as between the upper and lower bearing sleeves is, of course, that it is necessary for their open ends to face downwards, otherwise dirt would settle in them. As an additional precaution,



The throw-out gear for front-wheel-drive is housed in the nose-piece of the front axle



The front end of the silencer is rubber mounted and has a flexible connection to the exhaust down-pipe

it has been found necessary to fit a washer in the lower end of the upper bearing sleeve; this is because there is a relatively large clearance between the sleeve and the spring loaded end of the seal. This washer is clamped between the flange at the end of the screwed bush and the shoulder against which it bears.

The front suspension is almost identical with that of the rear, and most of the components, including the wishbone arms, are interchangeable. A swivel pin angle of 13½ deg and the castor angle 1½ deg have been adopted. The camber angle is 1½ deg and the toe in is ⅛ in. From the fully laden to the full bump position, there is a 2½ deg negative change in camber angle, and the track change is 1¾ in. The deflection to the fully laden position is 4·6 in; to the full bump position it is 7⅝ in.

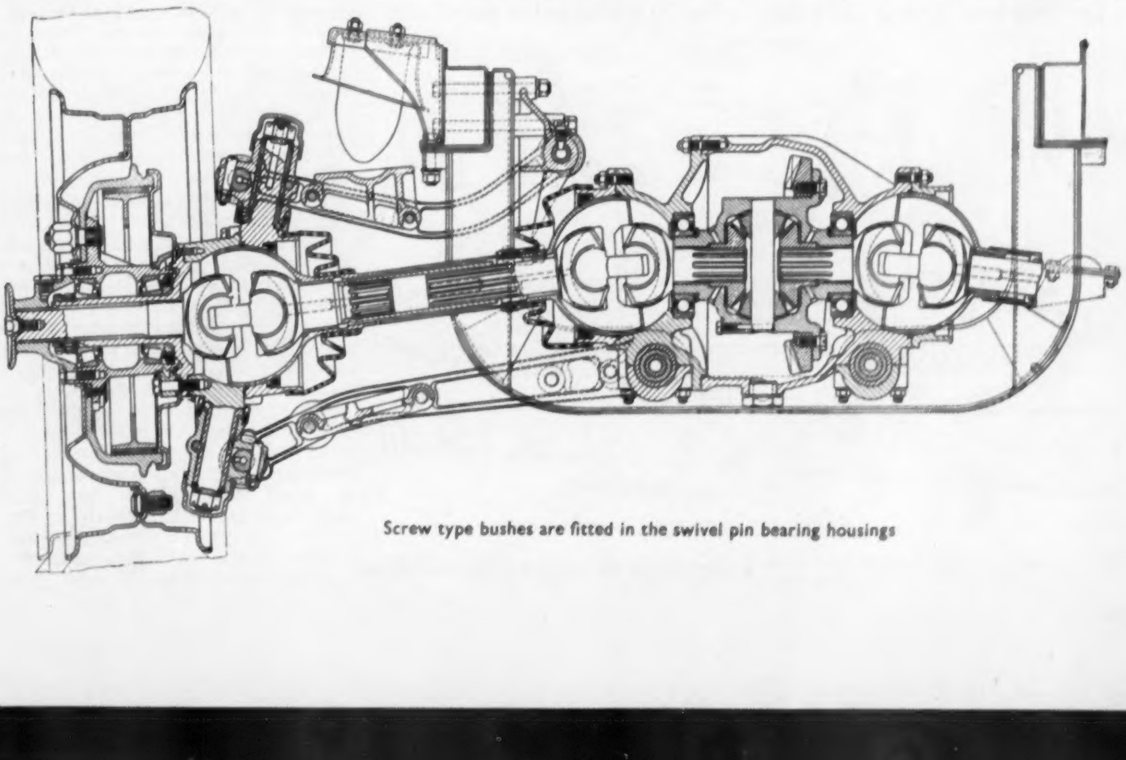
At both the front and the rear, the torsion bars are of En 45A and are 1·200 in diameter × 35$\frac{11}{16}$ in long. The bars are hardened, tempered and shot-peened. Their end fittings are of malleable cast iron B.S.310 grade 2. The attachment of the ends of the bar to the centre of the frame differs from that at the wishbones, which has already been described. All four bars are anchored on two brackets bolted beneath the centre of the cruciform frame. These brackets are of angle form and there is one for the two torsion bars each side of the suspension. One arm of each angle is bolted up to the gusset plate below the centre of the cruciform and the other is bolted to the side faces of the cruciform member. Two housings, or bosses, into which the 1¼ in diameter ends of the torsion bars are spigoted, are formed at the junction between the two arms of each angle. They are split in the manner of a bearing housing and cap, each of the caps being retained by four ⅜ in diameter studs and nuts. The spigot ends of the torsion bars are located in the housing by a circular section circlip in a groove round its bore and round the torsion bar end. This method of location permits the bar to float axially to a limited extent in the boss on the wishbone arm, to allow for expansion and contraction and for slight variations in the length of the frame.

Immediately outboard of the spigot ends, the torsion bars are increased in diameter to 1½ in and are splined to receive the En 17 adjustment arms. These arms extend upwards on each side of the cruciform. At their upper ends, forked eyes are incorporated to receive the end fittings on the adjustment rods. Each end fitting comprises a pin, which is carried in the forked eye, and two saddles, fitted one on each side of the pin. The threaded end of the ⅝ in diameter, adjustment rod is passed through holes in the saddles and the pin, and a lock nut is tightened against each saddle after the adjustment has been made. The other end of each rod has an eye formed in it, and is carried on a pin in a forked bracket bolted on top of the upper gusset plate at the centre of the cruciform.

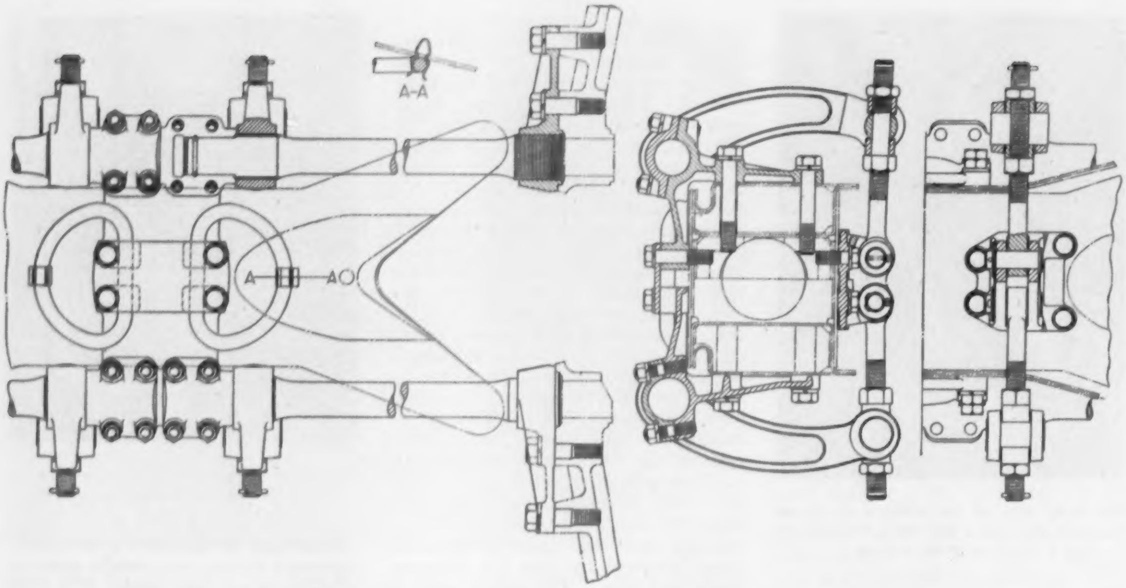
Two rubber-sandwich type mountings carry the tail end of the exhaust silencer

Steering

The rack-and-pinion steering unit is mounted on the front axle casing at a point immediately above the bearings that carry the crown wheel pinion. Thus, there is no possibility of relative movement between the steering box and the suspension, as there would be if the box were mounted on the frame. The housing for the rack and that for the pinion are integral and are of D.T.D.424. To allow for differential movement between the body and the steering box, and also to enable wider tolerances to be specified for assembly, a rubber bushed flexible coupling is interposed between the lower end of



Screw type bushes are fitted in the swivel pin bearing housings



All four torsion bars are anchored to a bracket on the centre portion of the cruciform frame

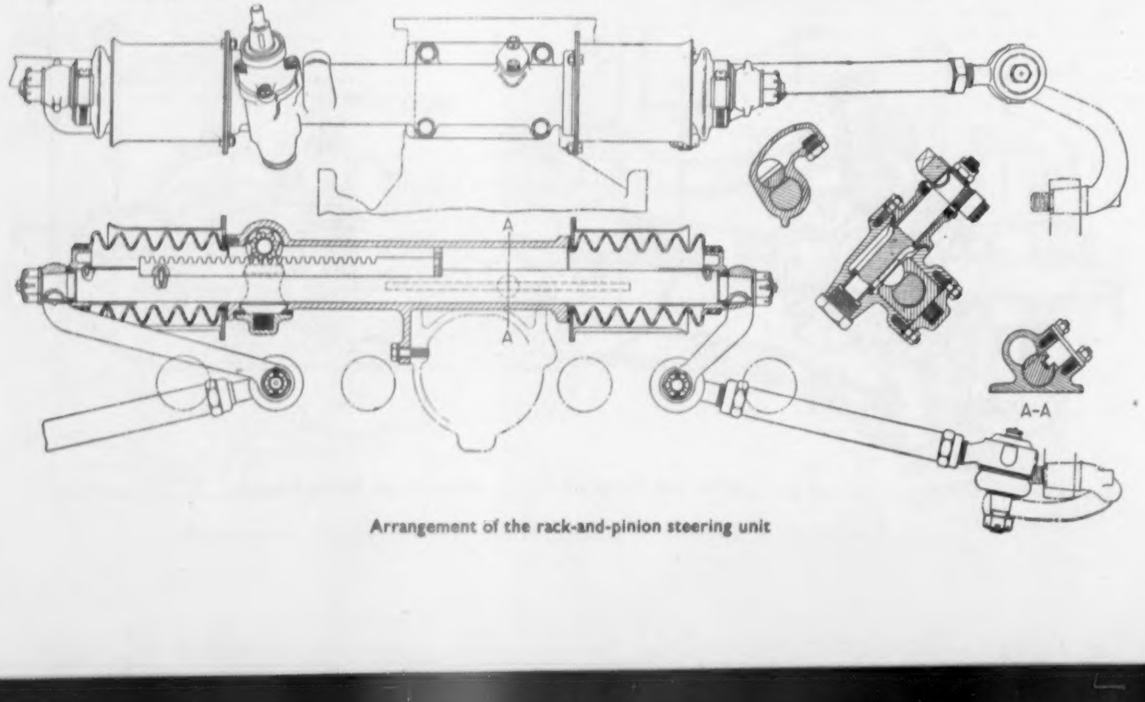
the steering column and the pinion spindle. This also has the advantage that it helps to absorb shocks and vibration transmitted through the steering system.

Two phosphor bronze bushes carry the En 39B, integral pinion and spindle. The upper one is about $\frac{11}{16}$ in inside diameter and is housed in a flanged cylindrical casting spigoted and bolted to the main casing. Above this bush, a lip type oil seal is carried in the end of the cylindrical casting. A cupped pressing, assembled on to the flexible coupling, enshrouds the seal housing. The flexible coupling is pulled on to a taper by a self-locking nut on the $\frac{7}{16}$ in diameter threaded end of the spindle. It is positively located against rotation relative to the spindle by a Woodruff key.

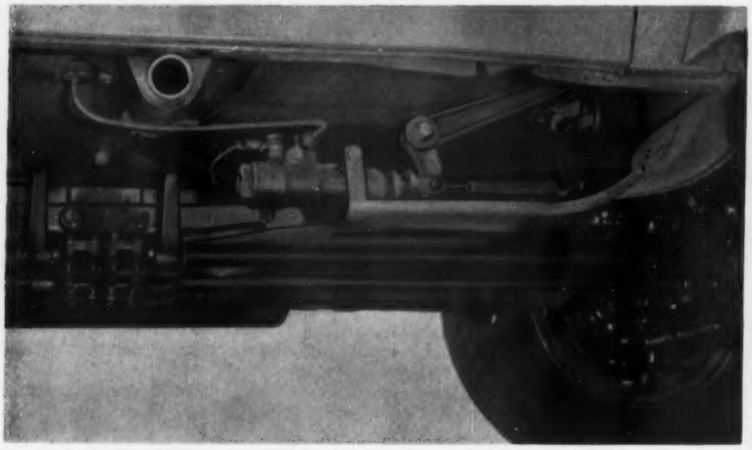
The lower bush is about $\frac{3}{4}$ in inside diameter; it is housed in a hole in the main casing. This hole is sealed at its lower end by a screwed-in plug. The inner ends of both bushes are flanged to form thrust faces and the pinion is carried between them. Two steel thrust washers, one at each end, are interposed between the bushes and the pinion.

A two-piece rack is employed. It comprises a $10\frac{1}{4}$ in long toothed component of semi-circular cross section mounted on a flat on a $26\frac{1}{4}$ in long circular section rod. Thus, if the teeth become worn after extended service, only a relatively small part of the unit needs to be replaced. The toothed portion is of En 9 and is induction hardened in the area where it is subject to the most wear. For the rod, En 17 steel is employed.

The flat is machined on the rod only to accommodate the toothed component, not to form a seating for it. Thrust from the pinion is taken directly through the toothed component of the rack to a phosphor bronze thrust block. This block is, in general, cylindrical in shape, and its axis is at right angles to and directly below that of the pinion. One end of the block is machined to form a semi-circular seating for the toothed component and the other bears against a thrust pad, of aluminium alloy, to S.A.E.306 specification, bolted up to the lower end of its housing in the casing. A coil spring in a hole in the centre of the pad ensures that the gear teeth mesh firmly together. If the wear between the two becomes excessive, it can be taken up by removing shims from between the pad and housing. A hole is machined through



Arrangement of the rack-and-pinion steering unit



The brake master cylinder is mounted on a bracket under the floor

the centre of the thrust block to clear the rod that carries the toothed component.

Lateral thrust on the rack is taken from the ends of the toothed component to the shoulders at the ends of the flat on the rod. A distance piece, which also serves to guide the toothed component in the casing, is interposed between the shoulder at one end and the toothed component, to which it is secured by a set screw. Near the other end, a peg is driven into a hole in the flat and registers in a clearance hole in the toothed component to make it impossible to assemble the two the wrong way round.

The rod bears directly in the casing, which is about 11$\frac{7}{8}$ in long. It is located against rotation by a tongued guide, which is inserted into a hole in the side of the casing to register in a $\frac{1}{4}$ in wide by 8$\frac{1}{2}$ in long slot in the rod. This guide is retained by a plate bolted over the outer end of the hole in which it is housed. Each end of the rod is shouldered and tapered to receive the keyed-on En 111, range T, arm to which the steering rod is attached. The arm is pulled on to the taper by a split-pinned nut on the $\frac{5}{8}$ in diameter end of the rod, and the centres of the ball joints on the ends of both arms are 3$\frac{19}{64}$ in from the axis of the rod.

Great care has been taken in the sealing of the ends of the unit. Circular end fittings and rubber sealing rings are interposed between the arms and the shoulders on the rod. Each of the two end fittings is located against rotation by a peg in the shaft. The sealing arrangement is completed by rubber gaiters. These gaiters are secured by a circlip to the circular end fittings and are clamped at their inner ends between a plate and the main casing. The gaiters are surrounded by steel sleeves that prevent stones from damaging them and also screen them from mud, which might build up between the corrugations until they become solid.

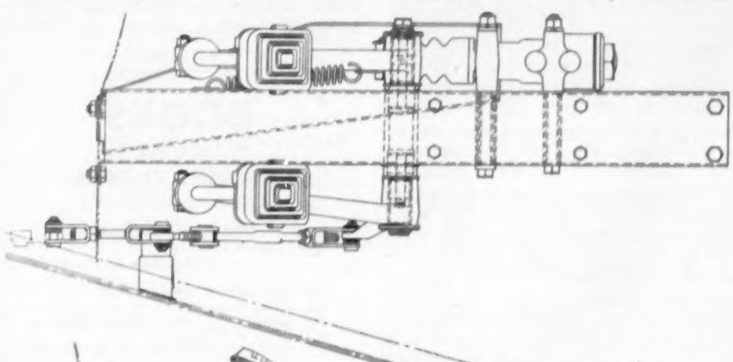
The steering rods are of tubular form, with screwed-in end fittings. Between the centres of these end fittings, the length of each rod is 11·83 in. The steering arms at the wheels have tapered ends and are positively located by Woodruff keys in the tapered holes into which they are fitted in the stub-axle carriers. On full lock, the angles of the two front wheels are 32 deg and 26 deg. This lock is obtained with 2$\frac{1}{2}$ turns of the 17 in diameter steering wheel. The turning circle is 35 ft diameter.

Brakes

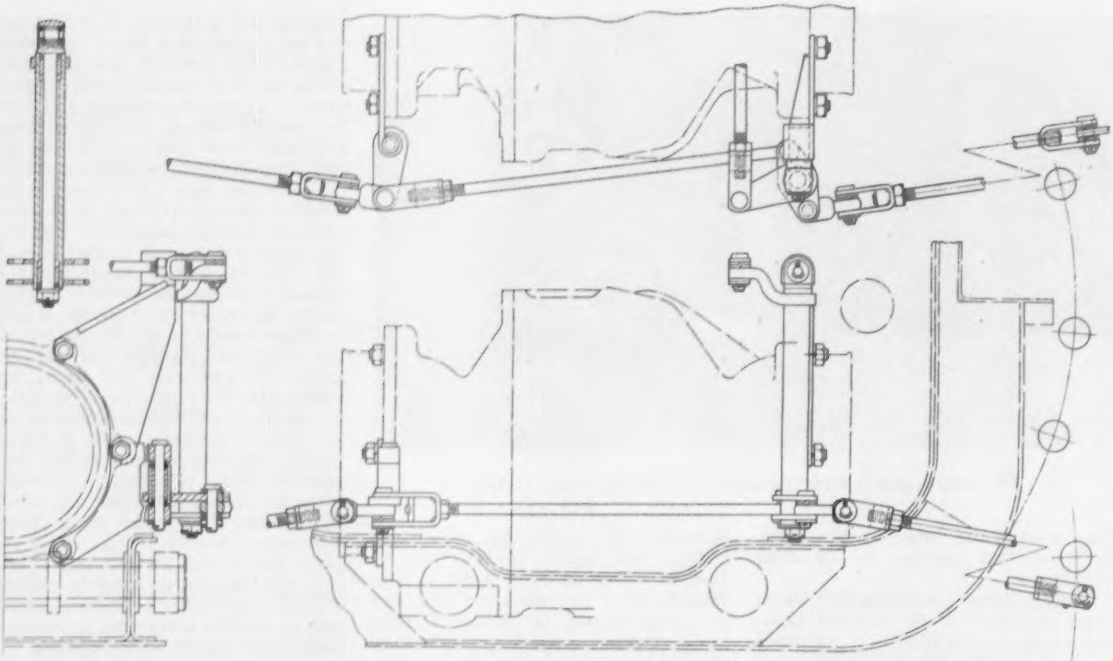
Girling hydraulic brakes are employed. At the front, the two leading shoe type of layout has been adopted, and at the rear, a one leading and one trailing shoe type is employed. The B.S.1452, grade 17, cast iron brake drums are 10 in diameter and the shoe width is 1$\frac{3}{4}$ in. At the front and rear, the friction linings are 9$\frac{7}{16}$ in long. The pedal lever ratio is 3·77:1 and the maximum available pedal travel is 6 in.

The brake and clutch pedals are mounted on a three-piece pivot comprising a central tube, with the two En 3 pivot pins copper brazed in each end. In this way an oil reservoir is formed between the pins. To distribute the oil to the bearings, a hole is drilled axially from one end of the assembly right through one pin and about half-way into the other, and radial holes distribute the lubricant to the spaces between the two bushes that are pressed into each end of the pedal bosses. By drilling the axial hole after assembly, only one drilling operation is needed instead of the two that would be required if the pins were drilled separately. Moreover, since only one pin is drilled through from end to end, only one screwed-in plug is needed. The oil reservoir is filled on assembly and no further lubrication is required throughout the life of the vehicle.

A circlip and plain washer on the outer end of each pivot pin retain each pedal. The two bushes in the pedal boss are $\frac{3}{8}$ in long by 0·7525-0·7535 in diameter and are spaced $\frac{1}{16}$ in apart. A rubber ring is inserted in each end of the boss that houses these bushes. These rings retain the lubricant and prevent the ingress of abrasive matter. A plain washer is interposed between the pedal boss and



An oil reservoir is formed by the tube, in the ends of which are brazed the pivot pins for the brake and clutch pedals



Layout of the handbrake control. The ends of the pivot bearings of the compensator are sealed with rubber rings, which prevent the entry of abrasive matter

the central tube of the pivot pin assembly. The whole assembly is carried by an inverted channel section bracket bolted up under the floor. A $1\frac{1}{4}$ in diameter hole is punched in each wall of the channel and the tube is passed through these holes and brazed-in.

Both pedals are of bell-crank lever form. The shorter arm of the brake pedal lever is connected to the master cylinder, which is mounted on the outer face of the channel that carries the pivot assembly. Two $\frac{3}{8}$ in diameter bolts secure the master cylinder; distance tubes are fitted between the walls of the channel and the two bolts are passed through them. The short arm on the clutch pedal lever is connected by an adjustable rod to another bell-crank pivoted on the forward end of the channel. Thence the motion is transmitted by another adjustable link to the lever on the clutch withdrawal mechanism.

The hand brake is of the lever type with a single sector, pawl and ratchet. It is mounted on the side of the cruciform member of the frame. Adjustable rods are employed to transmit the motion to the swinging lever type compensator mounted on the final drive casing. These control rods pass over the final drive casing, whereas the transverse rods from the compensator have to be connected to the brake at a much lower level. Therefore, the swinging compensator is of unusual form in that its vertical spindle axis is about $6\frac{3}{4}$ in long between the centre of the pin on which it swings and the centre of the lever to which the transverse rods are connected.

A tube about $6\frac{13}{16}$ in long by $\frac{15}{16}$ in outside diameter by $\frac{1}{2}$ in inside diameter carries the pivot bearing assembly for the compensator levers. At its upper end is welded the lever arm to which the rod from the hand brake control is connected. Two centrally pivoted levers are welded parallel to one another at its lower end to form forked connnections between which are pinned the ends of the transverse rods. The ends of the tube on which the levers are mounted are counterbored to take the $\frac{1}{2}$ in long by 0·565-0·566 in outside diameter by 0·4387-0·4397 in inside diameter bronze bushes, which are of the self-lubricating type, and rubber rings are inserted in the outer ends of the counter-bores to seal them against the ingress of abrasive matter. An eye is formed on the upper end of the pin on which the tube and lever assembly is pivoted. In it is a $\frac{3}{8}$ in long bronze bush of the self-lubricating type. This eye is carried on a pivot pin brazed to a bracket bolted to the final drive casing and it forms the axis about which the compensator swings.

This compensator is on the right-hand side of the final drive casing. As has already been mentioned, the connections to the transverse rods are pinned between the two levers at its lower end. The connection to the right-hand rod is of the universal type, to accommodate the swinging motion of the compensator as well as the vertical motion of the wheels, but the other end of the lever is pinned to an eye on the end of an adjustable rod connected to an idler lever pivoted on a pin mounted on a bracket on the left-hand side of the final drive casing. This idler lever is pivoted on a self-lubricating bush, sealed by rubber rings at each end in much the same way as those for the compensator. The universal joint to the transverse rod to the left-hand brake is also connected to this idler lever.

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

A SUBSIDIARY company, Churchill-Sturm Ltd., with offices at 27 Walnut Tree Walk, Kennington, London, S.E.11, has recently been formed by Charles Churchill and Co. Ltd., South Yardley, Birmingham, 25. This new company has acquired a licence to manufacture and sell the Sturm variable speed hydraulic drive in Great Britain and throughout the British Commonwealth. The unit will be marketed under the name Churchill-Sturm hydraulic drive. It is fully patented in this country and in many countries overseas, and has been used with great success in many industries on the Continent.

The Churchill-Sturm hydraulic drive, see Fig. 1, has an unusually wide range

Fig. 1. Hydraulic drive unit
(Churchill-Sturm Ltd.)

of speeds in combination with infinitely variable speed regulation. It gives constant torque up to about one-fifth maximum speed and constant horsepower throughout the remainder of the speed range. Furthermore, the unit is reversible, and the full range of speed and power is available in either direction of rotation. Because the unit is so designed that frictional losses are at a minimum, it has an unusually high efficiency, which is virtually constant under all normal operating conditions.

Essentially, the Churchill-Sturm hydraulic drive consists of an oil pump driven by a constant speed electric motor or other prime mover, and driving an oil motor which is connected to the output shaft. Both pump and motor are enclosed in a common casing that serves as a reservoir for oil. They are similar in construction, comprising a rotor fitted with sliding vanes that revolve inside a housing that is adjustable transversely inside the casing, so that the eccentricity of the housing relative to the rotor can be varied as required. Any desired speed within the range can be obtained merely by altering the positions of the pump and motor housings.

As the design of the pump and motor allows much higher oil pressures to be used than are normal for units of this type, the Churchill-Sturm hydraulic drive is remarkably compact. All moving parts are adequately lubricated automatically, so that wear is negligible and the unit requires practically no maintenance. In addition, as the unit is totally enclosed and operates in a closed circuit, there is no loss of oil. The only attention that is required is to change the oil at regular intervals.

This unit is available in a number of sizes ranging from 1·25 to 32 h.p. It can be supplied in various forms to suit the particular machine which it is to drive. Different types of control gear, including remote and automatic, are also available. In general, the unit can be adapted to suit the specific conditions of each application. It is particularly suitable for machinery that requires a reversible, infinitely variable speed drive which is compact, efficient and reliable.

Automatic driving centre

A lathe accessory designed to eliminate the driving plate and lathe dog is illustrated in Fig. 2. It is a product of Rotor Construction d'Appariels Scherler and Co., Zurich, Switzerland, and is designated the Rotor automatic driving centre. In operation, the work is set up between the live tailstock centre and the automatic driving centre which incorporates a spring-loaded centre and driving disc. When the workpiece is pressed against the driving disc, the spring-loaded centre is automatically locked and operates as a fixed centre to ensure concentric running under high operating pressure.

The self-aligning driving disc, with serrations on its face, automatically adjusts itself to the end of the workpiece if it has not been faced square to the axis. Variations in depth of workpiece centres are compensated for by the spring-loaded centre. Four

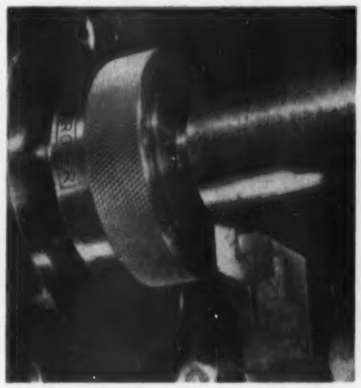


Fig. 2. Rotor automatic driving centre
(Insley Industrial Supply Co. Ltd.)

driving discs, which are easily interchangeable, are supplied with the automatic driving centre. They allow a wide range of diameters to be handled.

Another product of the same company is the Rotor rotating lathe centre designed to run true within 0·0001 in. The rotating centre is guided by its cone shaft and is provided with

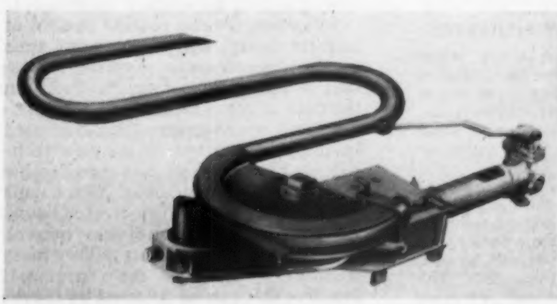


Fig. 3. "Senior" ratchet bending machine

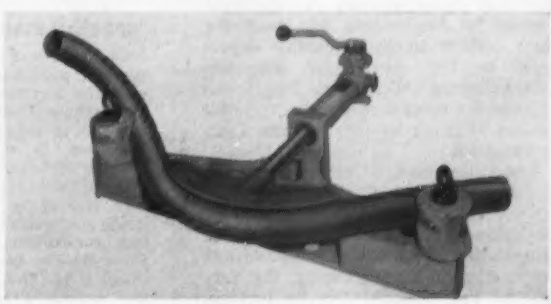


Fig. 4. Conduit former

(Chamberlain Industries Ltd.)

two adjustable roller bearings, with a ball thrust bearing to take the axial thrust. Insley Industrial Supply Co. Ltd., 21-22 Poland Street, London, W.1, are the agents for these accessories.

Pipe bending machines

For some time, Chamberlain Industries Ltd., Staffa Road, London, E.10, have considered that there is a need for a light, portable, hydraulic machine of reasonable price for bending non-ferrous tubes up to 3 in diameter. To meet this need, they have developed the "Senior" ratchet bending machine, illustrated in Fig. 3. It has a special type of bending head to fit on to a hand-operated self-contained pump to transfer the straight line thrust of the ram into a rotary movement that is necessary in order to bend light gauge tubes without causing wrinkling or flattening during the bending operation. Used in conjunction with correct formers, this machine is capable of bending ferrous and non-ferrous tubes of not less than 18 gauge wall thickness and up to 3 in diameter, to any angle up to 180 deg.

There are two main components: a frame, in which the hydraulic pump is located, and a bending arm which houses the adjustable back roller. The bending arm is rotated about the centre pin by means of an amply proportioned pin that is fitted to the pump ram. Formers are easily and quickly mounted on the attachment, and when used in conjunction with the special indicator plate that is fitted, automatically provide the correct lead angle for the type of tube to be bent. A pointer and angle plate, which are also incorporated, give the angle bend of a tube during any part of the operation.

Two of the main features of the machine are an automatic patent two-stage pump and the method of clamping the tube to the forming die, which allows for the production of close set bends with a minimum length of straight between the radii. Bends up to 180 deg can be made in one operation without removing the tube or adjustment of the die. The whole unit is self-contained and portable, and site fixing is not necessary.

Fig. 5. Abrasive band machine
(R. J. H. Tool and Equipment Co. Ltd.)

To extend the scope of another of their machines, Chamberlain Industries Ltd. have developed, as accessories, forming dies for bending 2½ in and 3 in electrical conduits without undue distortion. They can be bent cold and unloaded without any preparation. The machine, see Fig. 4, is readily portable and does not need foundation bolts. It is extremely light to operate, and the open type of frame offers many facilities when bends are made to template or where close bends in two planes are encountered. The unit develops a bending force of 8 tons.

As the two-stage action automatically ensures that the ram speed will adjust itself according to the load, unrestricted bending can be combined with maximum speed. The machine can be supplied with seven centre forming dies for bending electrical conduits from ¾ in to 3 in. With the aid of other attachments it is possible to bend various classes of tubes, both ferrous and non-ferrous.

Abrasive band machine

A recent addition to the range of abrasive band machines manufactured by the R.J.H. Tool and Equipment Co. Ltd., Artillery Street, Heckmondwike, Yorks., is illustrated in Fig. 5. This is a double-ended, self-contained upright model and is powered by a 4 h.p. totally enclosed motor. It incorporates the well-established Major backstand idler units and is designed to use standard 11 ft 4 in × 4 in abrasive belts.

The machine can be supplied with a motor running at 1,500 r.p.m., which, when 14 in diameter × 4 in face contact wheels are used, gives a band speed of 5,500 ft/min, suitable for heavy duty grinding and fettling of rough castings. Alternatively, the motor speed can be 3,000 r.p.m. for use with 10 in × 4 in contact wheels to give approximately a 7,500 ft/min band speed suitable for lighter duty grinding and polishing.

Free band strapping may also be effected above the contact wheel. The pedestal is cut away at the front to allow ample working clearance round the sides and below the contact wheel. Only 2 ft 6 in × 2 ft 6 in floor space is required, an important factor when room is limited. The overall height is 7 ft 9 in. Both tension and tracking handles are arranged within easy reach of the operator for quick and easy belt changing. The machine is strongly constructed of fabricated steel and has balanced pulleys mounted on sealed ball bearing spindles to give vibration-free working conditions. A push button starter is incorporated in the front of the pedestal.

ROUTINE METALLOGRAPHIC INSPECTION

FOLLOWING development work in the Metallurgy Section of Guest, Keen and Nettlefold Research Laboratories, a chemical method for brightening and polishing plain carbon steels, originally developed at the Armament Research Establishment, Woolwich, has been adopted for routine inspection of heat-treated forgings by Garringtons Ltd., Bromsgrove.

The component is immersed in the solution for 3 to 30 minutes: in the case of bulky components, a wall of plasticine is put on the surface to contain the solution. Since mechanical preparation is not necessary, the surface of the material can be directly examined, and the laborious cutting out of specimens and hand polishing of the surface that were hitherto necessary are completely eliminated. Moreover, the component is not damaged in any way.

As a result of this development, it is now possible to employ microscopic examination for the routine control of heat-treatment, and at the same time obtain a much more sensitive indication of the quality of the product. In the case of components for which conventional micro-examination was used in the past, the use of the new technique has led to a considerable increase in the number of components inspected from each heat-treated batch. Because of this, control has become more reliable and product quality more uniform. Among the structural features that can be detected by micro-examination are grain size and incomplete martensite formation.

SPECIAL SHOW NUMBERS

Our associated journal the "Autocar" will publish three issues dealing with the forthcoming International Motor Show. The first, a Show Guide, will give advance information about 1956 cars. It will be available on October 14th, price 1/–.

On October 21st a greatly enlarged number will be published at 2/–. It will give a stand-to-stand survey of cars, components, etc. Finally, the Show Review number will be published at 1/– on October 28th. This will provide a technical survey of design trends.

HARD-FACING EQUIPMENT

The Dewrance T.L.B. Powder Deposition Process

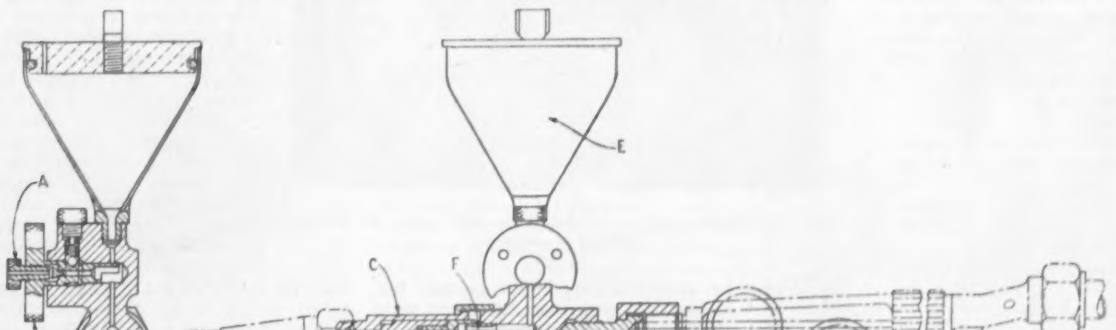
HARD-FACING alloys, resistant to heat, abrasion or corrosion, are commonly deposited on parts requiring protection by means of an oxy-acetylene flame. The alloy is supplied in the form of a filler rod and application is necessarily a two-hand operation demanding skill and close attention. It is difficult to ensure consistently uniform deposits of minimum thickness and it is usually necessary to make a second pass to float the alloy after initial deposition.

In the T.L.B. process developed by Dewrance and Co. Ltd., Great Dover Street, London, S.E.1, the hard alloy in finely powdered form is fed into the gas stream passing through the torch and is delivered at the welding tip. It is heated as it travels through the flame and is melted by the time it reaches the surface being treated, producing a true bond. Deposition is much smoother than is possible by the conventional filler-rod method and the depth of deposit can be readily controlled to reduce to a minimum both the quantity of material used and the amount of machining necessary to finish the work. Flux is not normally required and by suitable adjustment of the flame and pre-heating of the workpiece, a deposit up to $\frac{1}{8}$ in thick can be made consistently. This is carried out in a single run and without floating after deposition. The operation can be performed manually with one hand, but the flexibility of control and the ease of application make the process readily adaptable to automatic or semi-automatic operation.

Manual deposition is a simple, one-hand operation

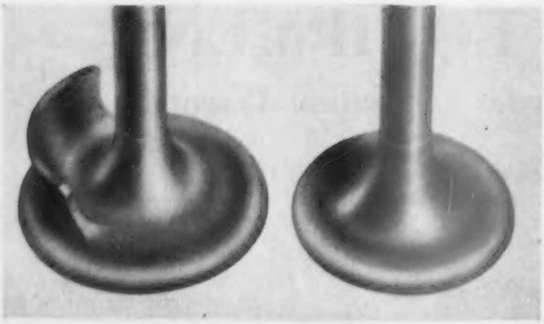
Only relatively simple and robust equipment is required to feed the powder into the gas stream. It comprises a hopper, valve and nozzle assembled as a unit and inserted in a standard B.O.C. type C.H. welding torch between the handpiece and the welding tip extension piece. It adds but little to the weight and need add nothing to the overall length of the torch. The body of the fitment is secured to the handpiece by means of an adaptor and a union nut, and at its downstream end supports the powder nozzle and its complementary brass sleeve enclosed in a housing. These three items are shouldered and so dimensioned that when the housing is drawn up by a union nut, the upstream end of the nozzle makes a coned seating in the body, and the coned downstream end is spaced from a coned face in the sleeve. To the end of the housing is attached the usual extension piece carrying the welding tip.

The gas stream flows through co-axial drillings in adaptor, body, powder nozzle and sleeve, and creates a depression in the coned annulus between the nozzle and its sleeve. This draws the powder from the hopper, past the control valve and through drillings in the body and the powder nozzle arranged parallel to the gas duct. Communication is by way of a hollow dowel pin, and from the annulus the powder is entrained in the gas stream. The control valve, which is operated by a large knurled knob, has three positions located by a spring-loaded ball detent. In the mid-position the powder is cut off and a turn of about 60 deg to left or right will give alternative rates of flow. Concentric with this valve is a small air bleed controlled by a needle valve. Normally this valve is closed, but should the powder surge into the flame, causing it to fluctuate, the valve

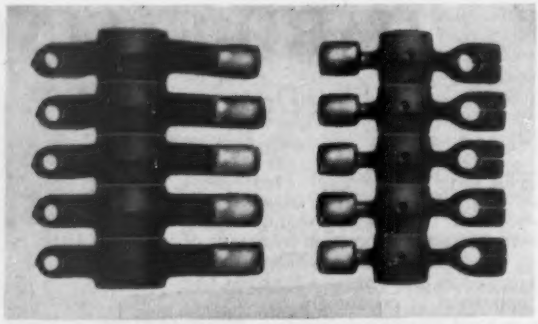


A—Air control valve; B—Powder feed control valve; C—Powder nozzle housing; D—Powder nozzle; E—Powder hopper; F—Orifice pin

Sectional arrangement of attachment fitted to a standard welding torch



Internal combustion engine valves faced with a nickel-boron-silicon alloy



Hard-faced ends on rocker arms for marine fuel injection pumps

is adjusted to admit a small flow of atmospheric air, which serves to promote a smooth feed of the powder.

Whilst the process was developed to use a wide range of Dewrance-Coast Metal alloys, it will handle equally well other powdered facing alloys, such as Colmonoy and Stellite, and also brasses, bronzes and copper. Representative of the Dewrance alloys are the 650/18 series, of chromium, cobalt, nickel, molybdenum and iron, which produce a facing of 385-430 D.P.N.; the 650/40 series, of nickel, chromium, tungsten and cobalt, giving 370-570 D.P.N.; and the 650/52 series, containing nickel, silicon and boron, for a hardness value of 660-750 D.P.N. It is essential that the powder, which is supplied in sealed tins, is kept free from moisture and covered to prevent the acquisition of dust or dirt. It is good practice to hold powder in a small oven at 80-100 deg C prior to use. All powders are passed through a 120-mesh screen in an Endecott or similar filter before being charged into the hopper and immediately covered by the plastics hopper lid.

Operation differs but little from normal welding practice. Ranges of four powder nozzles and seven welding tips are supplied with the torch, and selections of these are assembled in accordance with tabulated recommendations for facing operations of various widths and depths. Gas pressures are normally 10-15 lb/in^2 for both oxygen and acetylene. The work should be clean and free of oxides, mill-scale or oil. A machined surface is preferable, but when depositing a hard facing the most satisfactory bond is obtained if the surface is not too finely finished.

For the 650/18 and 650/40 series of powders, the work should be pre-heated with a neutral flame to bright red in the welding area, and then more acetylene introduced to give a carburizing flame with the white feather from two to three times the length of the blue cone. At the starting point, the base metal is brought to the sweating temperature and then the powder is turned on and deposition commenced. All oxides must be floated off as the welding proceeds, and to this end it is desirable to work with the blue cone of the flame just above the surface of the work. At the end of the run the powder is turned off, but the flame should continue to play on the work for a short period and then be slowly drawn away so that the molten pool of deposited material cools off from the inside outwards.

With the 650/52 series of powders, which have a lower melting point, a neutral flame is used and the work is pre-heated to a cherry red. As before, the work is locally brought to the sweating temperature, but the rate of deposition can be slightly quicker. The torch should be held with the nozzle tip about ¾ in from the work surface.

Semi-automatic machine for depositing hard alloys on annular facings or seatings

As stated earlier, the process, which is covered by patents in this country and abroad, is particularly suitable for automatic operation. The firm has built a number of special machines for semi-automatic operation on long production runs. That illustrated, for depositing annular facings on such items as steam valve discs, clutch members and brake discs up to 10 in diameter, may be regarded as typical. Such a machine is completely self-contained. The work is mounted on a circular table and rotated at appropriate speed by an electric motor driving through a variable speed gear and a worm gearing which is engaged or disengaged by a positive clutch. A ring of town gas jets is provided to pre-heat the work, and can be regulated to maintain the required temperature during the deposition of the facing and, if necessary, to stress-relieve the work on completion. To reduce the cycle time, the work may be pre-heated in a furnace and the gas jets used only to maintain operating temperature.

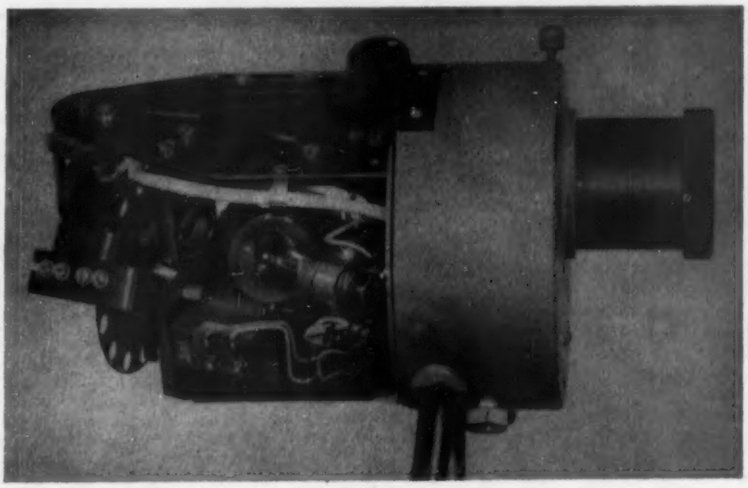
Two water-cooled welding torches, each fully and independently adjustable in respect of height, radial position, and angle relative to the work table, are mounted on a tubular framing which is reciprocated horizontally by a small electric motor. The drive is through an adjustable-throw crank mechanism so that the stroke of the torches can be suited to the width of the required facing. Each torch has its independently regulated gas and powder supply. Cycle time will vary from job to job under the influence of a number of factors. A typical time for depositing a facing approximately 10 in outside diameter, 1·0 in wide and ¼ in thick on a plain disc is 6 min.

THE MILLISCOPE PYROMETER

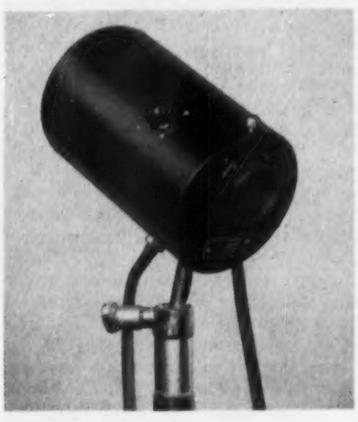
A Quick-response Instrument for Registering and Controlling Temperature Changes on Rapidly Moving Objects

IN optical pyrometers of the incandescent lamp type, the filament of the comparison lamp is located in the path of the radiation emanating from the heated object under examination. The colour of the filament is compared by the eye with that of the radiation, and by regulating the lamp current an incandescent colour is produced to match the radiation colour. The current required by the lamp is then used as a measure of the temperature of the object.

Temperature changes commonly occur so slowly that such instruments can register temperatures without material error arising from time lag. The accuracy of the reading will, of course, be dependent on the skill and experience of the operator. However, the human eye is but an indifferent instrument for such purposes. Its sensitivity to colour may vary with changes in either physical, emotional or environmental conditions, and it is highly susceptible to fatigue. The well known phenomenon of "persistence of vision," so valuable in some respects, is a definite handicap in the rapid assessment of changing values.



Working head with cover removed, revealing comparison lamp and its lens, the lead sulphide cell, and the chopper disc and its motor



External view of the working head. The knob controls the hinged focusing mirror

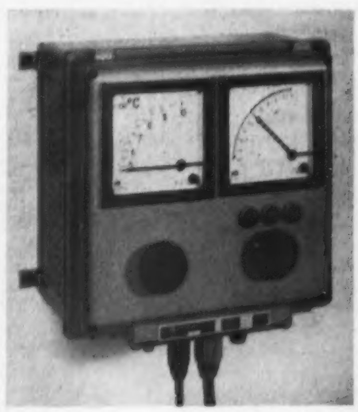
Where relatively rapid temperature rises are encountered, as when heating parts with an oxy-gas flame or by induction, the usefulness of the instrument is seriously reduced. The higher the rate of temperature rise, the more the registered temperature lags behind the temperature attained by the heated part. Should the rate of temperature rise exceed a magnitude of about 80 deg C per second, the instrument ceases to be of practical value.

Developed initially for the observation and control of flame-hardening operations by an oxy-gas process, the Milliscope can follow such rapid temperature rises without appreciable time lag. It employs the principle of the incandescent lamp pyrometer by comparing the radiations of two bodies, the heated object and the filament of a calibrated incandescent lamp, within a certain wave band. The receiving element is a lead sulphide cell and thus the human eye, a potential source of inconsistency and error in the ordinary radiation pyrometer, is eliminated. Since the transformation of radiation energy into electrical energy in such a cell is virtually instantaneous, the time lag is dependent solely on the inertia of the indicating instrument and can be held within reasonable, practical limits.

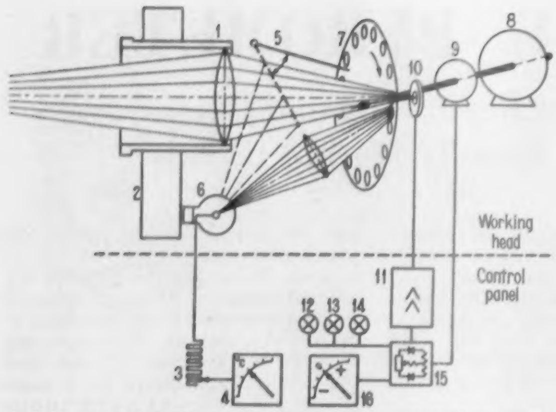
An incidental advantage of the lead sulphide cell for this purpose is that it is responsive to the infra-red band of the spectrum. Consequently, it is not influenced by the high temperature flames, often blue in colour, which may be around or adjacent to the heated workpiece.

In the Milliscope the paths of the two radiations are arranged optically separate, and comparison is effected in the following manner. A chopper disc is mounted in front of the lead sulphide cell and driven by a small synchronous motor. The location of the disc, and the pitching of its apertures, is such that radiations from the object and from the lamp filament are projected alternately on the cell. When both radiations are of equal intensity, the cell is influenced continuously to the same extent and produces a direct current voltage. Conversely, when the radiations are unequal, an alternating current voltage is produced.

By using an alternating current amplifier, this alternating voltage could be registered on a suitable instrument. The greater the difference between the intensities of the two radiations, the higher would be the voltage, but there would be no indication as to which radiation was the stronger. When the radiations became equal, the voltage

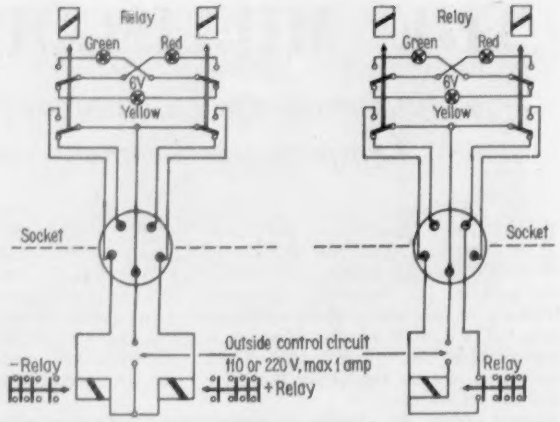


Control panel with temperature-setting indicator on left and zero instrument on right



1—Telescoping lens; 2—Water jacket; 3—Rheostat for control of lamp filament temperature; 4—Temperature pre-setting and indicating instrument; 5—Hinged mirror; 6—Comparison lamp; 7—Chopper disc; 8—Motor; 9—Auxiliary generator; 10—Lead sulphide cell; 11—Amplifier; 12, 13, 14—Pilot lamps; 15—Phase bridge and relay; 16—Zero instrument.

Schematic arrangement of instrument and control panel



Wiring diagrams for typical automatic controls of heating operations. Left: Control of feed rate for progressive spin-hardening on a vertical or horizontal flame-hardening machine. Right: A constant temperature control for normalizing butt-welded seams

would disappear. Accordingly, in order to determine which is the stronger of the two radiations, a zero instrument sensitive to phase is incorporated in the circuit. The necessary auxiliary current is supplied by a small alternating current generator driven by the chopper-disc motor.

This generator has the same number of poles as there are apertures in the chopper disc and, therefore, generates a current of the same frequency as that of the disc. The zero instrument sensitive to phase consists of a bridge arrangement in which a direct current instrument is connected to a push-pull output rectifier. When the alternating current produced in the cell equals zero, the instrument will register zero regardless of the magnitude of the auxiliary current. A positive value will be indicated when the auxiliary voltage is in phase with the cell voltage and a negative value will be shown when the two voltages are displaced 180 deg.

On the control panel, the knob on the left enables the rheostat, controlling the current to produce the required temperature in the filament of the comparison lamp, to be regulated. The desired temperature is simultaneously indicated on the dial immediately above the knob. To the right is the zero instrument, the pointer of which commences to move about 25 deg C before the workpiece reaches the required temperature and registers zero when it is attained.

A built-in magnetic voltage regulator compensates for voltage fluctuations of +10 per cent to −15 per cent, reducing them to less than ±1 per cent.

The working head of the Milliscope is enclosed in a plain cylindrical casing $5\frac{5}{16}$ in diameter and $8\frac{7}{8}$ in long, and weighs about 15 lb. From the water-jacketed front face projects the telescoping lens by which radiation from the heated work is focused on the lead sulphide cell. As an aid to setting up the head and focusing on the work, a hinged mirror is provided within the casing. By means of a two-position knob control, it can be swung from its normal inoperative position of stowage to a plane behind the lens where it reflects light from the comparison lamp back through the lens to project a bright spot of light on to the work.

By visual observation of the zero instrument, it is possible to control heating operations manually for such processes as forging, hardening and tempering. Since temperatures can be measured so rapidly and so precisely, however, it is an obvious step to introduce automatic controls. This can be quite simply effected by connecting the output of the phase bridge to relays, as well as to the zero instrument. These relays operate the green, yellow and red pilot lights, and also the switching contacts for the external control circuit. Two such wiring arrangements are illustrated. In one, controlling the feed rate of a flame hardening machine, two relays are provided. The minus relay is energized to reduce the feed when the work temperature falls below the pre-set value, while the plus relay is actuated to increase the feed should the work temperature exceed the desired temperature. The other example shows a single relay arranged to maintain a constant temperature in a normalizing operation on butt-welded seams. It shuts off the heating when the work reaches the pre-set temperature, and cuts in the heating as the work falls below that temperature.

Seven standard instruments are available. These cover different optical and temperature ranges at different target distances, and different target areas for the indication or control of temperature up to 1,800 deg C.

The Milliscope is produced by the Frankfurt firm of instrument makers, Hartmann and Braun, solely for Paul Ferd. Peddinghaus, of Gevelsburg i.W., Germany, whose sole representative in Britain is Surfard Ltd., 2 Victoria Street, London, S.W.1.

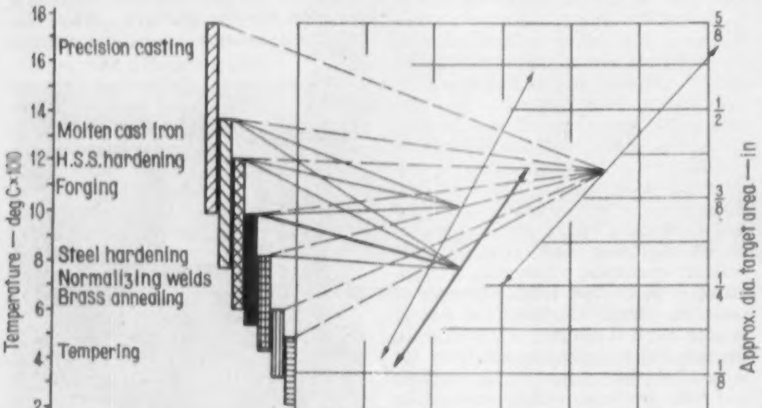


Chart showing available temperature ranges, target distances and target areas of seven standard models

AUTOMATIC MACHINE LOADING

An Interesting Development with Gisholt Machines

USUALLY, automatic machine loading is considered to be economically suitable only for long run production. It has, however, been applied with success to the batch production of electric motor frames in a two-operation sequence on Gisholt lathes. Six different electric motor frame sizes are handled in batches.

The first operation is carried out on a Gisholt No. 12 hydraulic automatic lathe equipped with a hydraulically operated special boring carriage, with rapid approach, infinitely variable forward feed and automatic rapid return. The length of feed is manually adjustable to suit either the long or short stacks of laminations. In addition, the machine is equipped with a Speed Variator drive unit, which permits working at the correct surface speed for the complete machining operation on all six different sizes. The machine for the first operation is illustrated in Fig. 1.

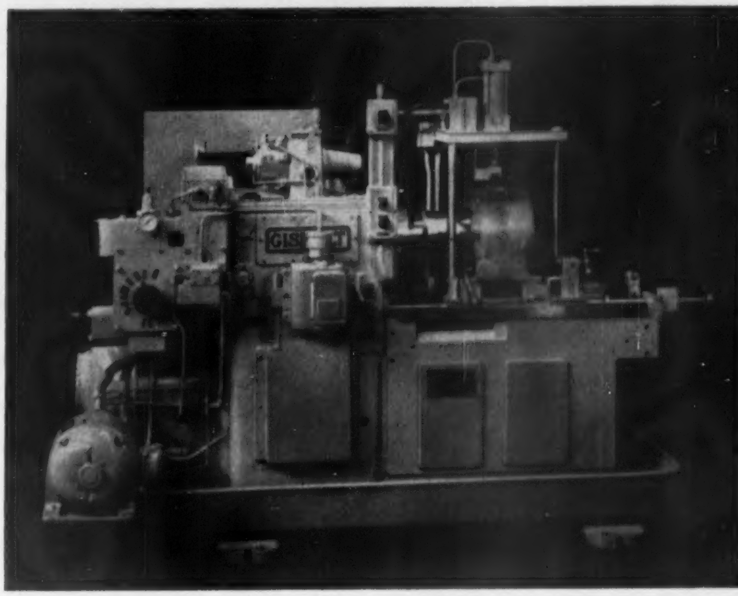


Fig. 1. Gisholt No. 12 hydraulic automatic lathe equipped for the first operation on electric motor frames

Before the frame comes to this machine, the height is established, since the feet are previously milled and drilled. When the work is delivered by conveyor, the operator slides it on to the machine carriage table. A guide rail assists rough location. To locate the frame accurately for machining, spring-loaded studs are manually raised to enter holes in the two frame feet nearest the headstock. Once the frame is located, the overhead clamp may be actuated. As soon as the frame is clamped, the electrical circuit is cleared and the machine motor can be started. The clamp and motor are push button controlled, and the arrangement of the push buttons is such that both of the operator's hands are occupied and out of the clamping zone. Traverse and feed cycle are started by raising the regular starting control lever.

When the cycle is started, the frame traverses to position and then feeds forward to engage a special rotating cutter head mounted on the machine spindle for boring the stator laminations. Angular ports in the rotating cutter head permit vacuum removal of the chips through the back of the spindle. When boring is completed, the spindle stops and the frame moves with reverse traverse to the unloading and loading position. As was stated earlier, six different frame sizes are handled, and only minimum change-over time is required. Floor-to-floor time for some sizes is as low as 1·1 minutes.

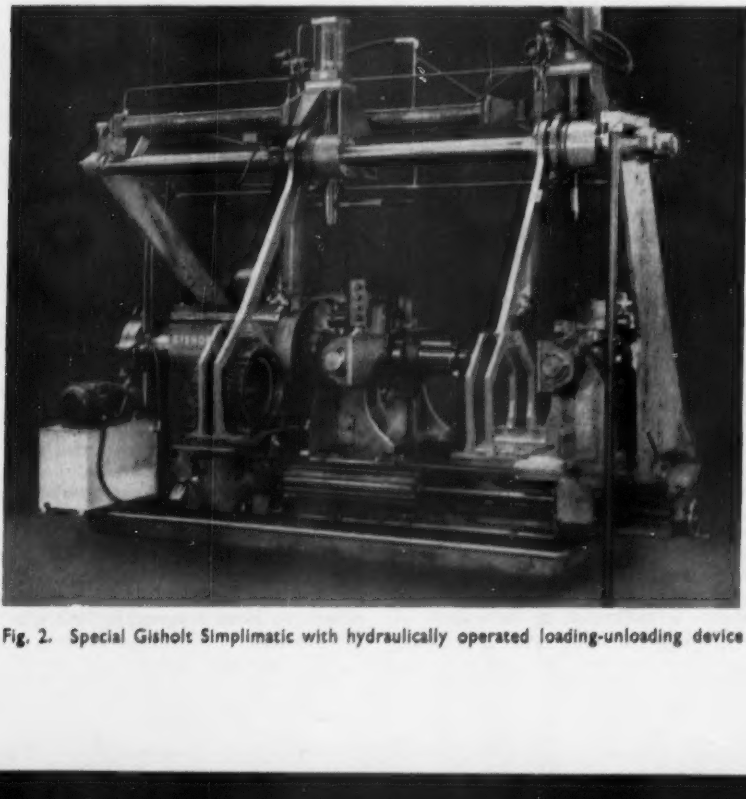


Fig. 2. Special Gisholt Simplimatic with hydraulically operated loading-unloading device

From the first operation the motor frames are taken by conveyor to a special Gisholt Simplimatic automatic lathe, see Fig. 2. This machine is equipped with two vertical slide housings, each arranged for two facing slides and two boring slides. One finish facing slide and one finish boring slide are mounted on each slide. The second operation machining consists in finish boring and facing rabbet and shield fits at both ends of the motor frame.

Fig. 2 also shows the special overhead mounted, hydraulically operated loading-unloading device that has been designed to increase production by reducing handling time. In the illus-

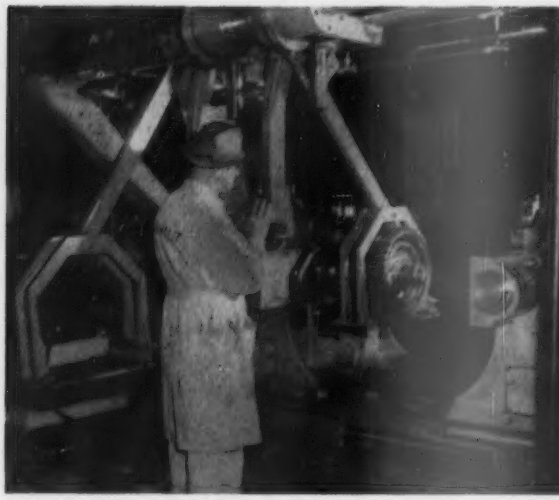


Fig. 3. The unloading cradle picking up a completed frame

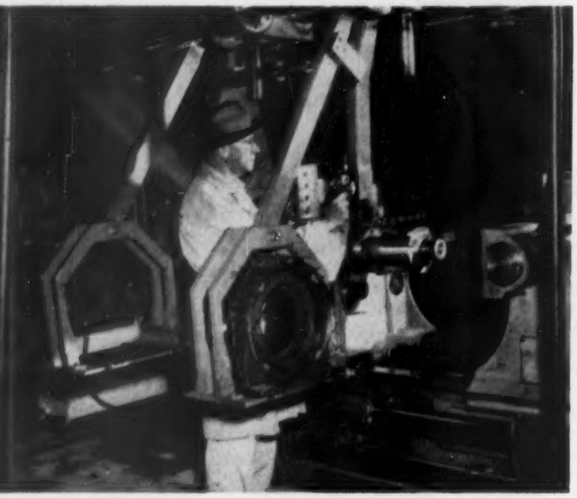


Fig. 4. Completed frame swung clear of the machine

tration, a part for the second operation, taken from the conveyor, is shown in the loading cradle at the left. In actual operation, there would also be a completed part in the unloading cradle at the right, ready to be transferred to the conveyor at any time during the next machining cycle.

In operation, the loading cradle is moved to the right, then swings to bring the work into loading position between the slide housings, followed by longitudinal movement to the left to place the work on an expanding mandrel for chucking. Location is from the edge of the frame with a manually operated swinging locating stop. The loading cradle then moves clear of the work and the machine, and the left-hand traversing slide housing moves in to support the mandrel. Tools on the rear slides of both slide housings finish face both outer faces, while tools on the lower boring heads finish bore and chamfer the rabbet diameters.

When machining is completed, the left-hand traversing slide housing retracts, and the expanding mandrel releases as the operator actuates the air valve to swing the unloading cradle into position to pick up and remove the completed part. Fig. 3 shows the unloading cradle after it has moved in between the slide housings and towards the spindle to pick up the completed part, while Fig. 4 shows the unloading cradle after it has moved out with the workpiece and cleared the machine ready for the next cycle.

Since this machine is to be used on relatively small batch production, special provision has been made for quick change-over from one frame size to another. For example, the loading cradle has removable shoes so that accurate location can be quickly obtained for any size of frame. Change-over for different bore sizes is easily and quickly effected by changing sleeves on the expanding mandrel, and a manually operated crank controls longitudinal positioning of the outer vertical slide housing for different frame sizes. Finally, to facilitate tool setting, dial indicators are mounted as part of the tooling. Burton, Griffiths and Co. Ltd., Mackadown Lane, Kitts Green, Birmingham, 33, are the British agents for Gisholt machines.

CLOSE LIMIT GRINDING

A Development on Scrivener Centreless Machines

MODERN high pressure hydraulic systems, especially those used in precision engineering, make exacting demands upon the grinding machines used for the finishing operations, since there are certain components, such as valves and pistons, which must not allow the slightest oil leak, even at the highest operating temperatures. When a centreless grinding machine is used for finishing components of this character, extreme accuracy rather than high production rate is the primary essential. Although these hydraulic components may be plain cylindrical pieces which would in normal practice be ground by the straight-through process, experience has shown that certain classes of components are best finished by the plunge-grinding method.

As is well known, when a centreless grinding machine is used for plunge-grinding, instead of passing the work straight through between two fixed wheels, the control wheel and work are slightly withdrawn from the grinding wheel at the commencement of the operation; the operator advances the control wheel and work towards the grinding wheel by pulling forward a lever until contact is made with a dead stop which gives the finished size. Any change in size that may be required is obtained by adjustment of the initial position of the control wheel.

It is, however, a feature of most centreless grinding machines that the final adjustment of the wheel cannot be readily obtained with the high degree of accuracy that is essential for high-pressure hydraulic valves. An important advance in this respect has recently been made by Arthur Scrivener Ltd., Tyburn Road, Birmingham. It consists in fitting a plunge-grinding micrometer stop, which is calibrated in increments of 0·00001 in, to allow parts for high-pressure operation to be ground to the required tolerance so that necessity for subsequent lapping is completely eliminated.

Workpieces requiring a stock removal of 0·005 in can be ground on this No. 1 Scrivener centreless grinding machine, equipped with the plunge-grinding micrometer stop, in five or six passes, according to the class of finish that is required. The first cut removes 0·004 in, and the final cut 0·0001 in to give a final tolerance of 0·000025 in for size, roundness and parallelism. Work of this small size naturally presents a certain difficulty in loading between the closely-spaced wheels of a centreless machine used for plunge-grinding, because of the danger that the operator's hand may be caught by the grinding wheel. To obviate this, a hand push-feed mechanism has been designed for use with this machine.

COATED ENGINE VALVES

General Motors' Aldip Process

J. Anthony Edwards

IN the automobile industry, where costs and engine performance are all-important, materials engineering has constantly to keep pace with these factors.

The Research Laboratories Division of the General Motors Corporation has recently devised a new aluminium coating process for engine valves, called "Aldip," which prolongs their life considerably. Details of the process were given before the Society of Automotive Engineers by R. F. Thomson and D. K. Hanink of General Motors Research Laboratories, E. B. Etchells of Chevrolet, and K. B. Valentine of Pontiac.

It is becoming increasingly important to improve valve durability because over the past 25 years there have been several long-term trends affecting this problem such as:

1. Increased car life. Cars scrapped in 1951 covered nearly five times the mileage, and ran over twice as long in years than those scrapped in 1925.
2. Increased car speed. The typical average car road speed today is about the maximum attainable 25 years ago.
3. Increased engine temperature. Increased volumetric efficiency in present high output engines has resulted in higher operating temperatures. Present-day valves open sooner and remain open longer.
4. Increased valve stresses. Higher lift cams and stronger springs with higher initial loads and higher spring rates have increased the stresses on valves.

Valves may fail for a variety of reasons. One of these is failure to seal due either to local removal of metal, such as in "guttering," or more general metal removal which causes loss of compression. The latter type of failure results from a complex series of variables which includes compression ratio, fuel, lubricant, temperature, deposits, and mechanical factors.

TABLE I.—Dynamometer test results

Cylinder No.	Type of valve	Running hours	Remarks
1	Aldip-coated XB	587	No failure
3	Aldip-coated XB	587	No failure
5	Aldip-coated XB	587	No failure
2	Plain XB	587	Failed—burned
4	Plain XB	144	Failed—burned
4	Plain XB (replacement)	377	Failed—burned
4	Plain XB (2nd replacement)	66	No failure
6	Plain XB	427	Failed—burned
6	Plain XB (replacement)	160	No failure

There are several solutions to increasing valve life, but one of the major considerations has been to try to resolve the problem by using the lowest possible strategic material content. Aluminium has long been added to the surface of iron to improve its high temperature oxidation resistance, especially in the presence of sulphur-containing gases. Typical applications are in molten bath furnaces for salt, cyanide and lead, bolts for use at temperatures up to 1,400 deg F, tubes for air heaters, and furnace parts to be used at temperatures up to 1,400 deg F.

Investigators have for some time been aware of the potential advantages of coating poppet valves with aluminium and many techniques have been tried, but were not commercially successful for a variety of reasons. Some methods failed because the alloy layer was too thick, causing "crazing" and "spalling" of the coating. No commercial process was available which would produce a smooth coating so that further finishing was not required. Other coating techniques failed because the bath could not be kept continually and economically operative. With the development at General Motors Research Laboratories of the Aldip process, which was not subject to these shortcomings, the process was tried on automobile valves to see if appreciable advantages could be obtained.

One of the lower strategic alloy materials which has been widely used as an exhaust valve steel is that known as Silchrome XB, the nominal analysis of which is carbon 0·60-0·85 per cent; manganese 0·20-0·60 per cent; silicon 1·25-2·75 per cent; chromium 19·00-23·00 per cent; nickel 1·00-2·00 per cent. This steel has many desirable attributes of a good valve steel and in some engines its only shortcoming was a failure to produce a sufficiently long life owing to failure to seal at the seat; burning or guttering was thus often caused. Intake valves are generally made of a low alloy material, such as GM 8440, 3140, 8640, etc. The analysis of GM 8440 to be described in the article is: carbon 0·35-0·45 per cent; manganese 0·25-0·40 per cent; silicon 3·60-4·20 per cent; chromium 1·85-2·50 per cent.

While these materials seem to have all the properties required for satisfactory intake valve performance, they occasionally become stuck open and exposed to hot gases because of deposits. This causes premature removal of metal from the valve head and loss of pressure in the cylinder head.

Valves are processed by the Aldip method after the seats have been finish machined and the stems roughly ground. Essentially, the process consists in dipping about 1 in of the head end of the valve into a molten flux bath, followed by immersion in molten aluminium. Subsequently, the valve is put in a fixture where compressed air is used to remove the surplus molten aluminium from the valve seat.

It has been found in subsequent development work that the processing costs may be appreciably reduced and the production rate increased by applying the aluminium to the valves before immersion in the molten salt bath. This may be done in a number of ways, and the following methods of application have been found to be successful.

1. As a paste by mixing aluminium powder with a suitable binder.
2. By metal spraying on to the article to be coated.
3. In the form of aluminium powder

TABLE II.—Field tests on two-ton trucks

Engine No.	Mileage at inspection	Number of burned valves	Type of valve
1	9,568	1	Uncoated
2	10,027	1	Uncoated
3	13,047	0	—
4	30,497	1	Uncoated
5	32,498	0	—
6	35,040	0	—
7	45,697	1	Uncoated
8	55,852	0	—

TABLE III.—Controlled accelerated car performance test summary. Exhaust valves

Type of valves	Number of engines tested	Number of burned valves	Total mileage all engines	Earliest failure, miles	Maximum life without failure*
Aldip	10	3	286,162	14,203	51,040
Uncoated	8	21	157,295	3,145	37,909

*Could be operated after this mileage.

TABLE IV.—Controlled accelerated car performance test summary. Intake valves

Type of valves	Number of engines tested	Number of burned valves	Total mileage all engines	Earliest failure, miles	Maximum life without failure*
Aldip	10	0	286,162	None	51,040
Uncoated	9	37	165,189	2,695	37,909

*Could be operated after this mileage.

suspended in a suitable vehicle which is sprayed upon the article to be coated.

4. In the form of a washer or as a wire ring set on the valve head.

In the process, the valves are first cleaned by vapour degreasing prior to being placed in a fixture which holds eight valves. The fixture is then placed on a conveyor rack and the head ends of the valves are automatically immersed in molten flux at 1,325 deg F. They are then automatically conveyed through a furnace and arrive above a pot containing aluminium. The pot rises, immersing the valve heads in aluminium for five seconds. The aluminium pot is lowered and the valves are immediately removed from the furnace and fixed in a blow-off machine where compressed air removes surplus aluminium from the valve seat face.

The nominal composition of the fluxing salt used is 40 per cent sodium chloride, 40 per cent potassium chloride, 10 per cent sodium aluminium fluoride and 10 per cent aluminium fluoride. Commerical 25 aluminium is used. These methods lend themselves to automatic processing which greatly reduces costs. The coating has been applied to an appreciable number of different valve steels, but the greatest amount of field data has been accumulated on the two steels described above.

Considerable test experience has been obtained by both dynamometer testing and car testing for several different engines manufactured by General Motors. Dynamometer data were obtained on one engine by the Ethyl Corporation. The test cycle was an accelerated valve durability schedule. Criterion for failure for any single valve was based on a continuing 25-lb cylinder compression loss during engine operation over a 24-hour period. The engines were run with three Aldip-coated and three uncoated exhaust valves. In these three tests, only two of the nine Aldip-coated valves failed, whereas nine original and eight replacement uncoated valves failed. The results of another typical test run under the same conditions with Aldip-coated valves in cylinders 1, 3 and 5, and plain valves in 2, 4 and 6, are shown in Table I.

In some field tests on heavy-duty 2-ton trucks in which each engine was equipped with a mixed set of valves, half Aldip-coated and half plain valves, all engines were equipped with valve rotators. It is significant that all four "burned" valves were uncoated and none of the Aldip-coated valves failed in this group. The results are given in Table II.

The beneficial effect of the Aldip-coating treatment was also observed in passenger car engines. Test cars, some with engines fitted with Aldip-coated valves and others with engines fitted with plain valves, were operated under accelerated test conditions for various mileages up to 51,040 miles. Summaries of the data obtained for the exhaust and intake valves are shown in Tables III and IV.

The Pontiac Motor Division of General Motors Corporation has operated an Aldip installation for about a year, for coating valves. Chevrolet Motor Division also has an installation now in operation.

GRAPHITED SURFACES

A New American Coating Process

AFTER extensive research, the American Electrofilm Corporation has recently developed a means of applying a stable graphite film to almost any kind of surface. The process, designated the "Electrofilm graphite process," has been developed to allow full advantage to be taken of the highly desirable lubrication characteristics of graphite. As supplied by this process, the stable film is extremely thin (0·00015 to 0·0005 in.). The adhesion to the treated surface is high, and on metal and most other surfaces, sufficient diffusion of the graphite into the surface is obtained to ensure presence of a graphite coating even when the external coating is apparently removed.

Following the normal surface preparation, the process is applied by spraying or dipping. It has been used successfully on metals, plastics, rubber and ceramics. For most purposes, plating is not required, but if necessary, Electrofilm graphite can be applied to plated parts. The film is stable and provides the desired surface characteristics over temperatures ranging from −120 deg F up. Some variations have been successfully tested to 2,000 deg F.

SHOW REVIEW NUMBER

The extra issue of the "Automobile Engineer" will be published as usual in connection with the London Motor Exhibition. It will constitute a critical review of the more interesting exhibits, including coachwork, and will have numerous illustrations of special features and design characteristics.

Publication of this issue will follow the ordinary November issue, which will appear on November 2nd. The Show issue will be available on November 23rd, and can be obtained by order from newsagents throughout the United Kingdom, price 3s. 6d. net. Readers are reminded that it is still necessary to make arrangements with the newsagent to make certain that a copy is secured.

In actual service most promising results have been obtained in the reduction of friction and wear on engine components. In one of the early applications, pistons, piston rings and bearing inserts of an aircraft auxiliary power unit were graphited. Before treatment, the normal cylinder head temperature for this engine after a five-hour run-in at reduced speed was 500 deg F. A similiar engine with graphite treated components was put into operation at rated speed and full load without run-in. The maximum head temperature was reduced to 450 deg F and there was no increase in oil temperature.

In a effort to reduce the severe scoring which had been experienced during initial running, graphite coated pistons were installed in a test engine at an automobile plant. Testing was conducted with elevated oil and water temperatures, but even after 2,000 hours of operation there was no evidence of scoring.

GUDGEON PIN HOLE DRILL

An Interesting Special Purpose Machine

A MACHINE for drilling the gudgeon pin holes in pistons for Humber engines has recently been developed by Adcock and Shipley Ltd., Ash Street, Leicester. The machine, which is shown in Fig. 1, is of the horizontally-opposed type with two multi-spindle heads, and with a two-station fixture designed to enable two components to be drilled at one station while loading and unloading are carried out at the other station.

In the design of this machine, special attention has been given to the elimination of operational fatigue and to the safeguarding of the operator while retaining a high production rate. Manual operations have been reduced to a minimum by the use of hydraulic power for clamping the components and indexing the fixture. The indexing of the components in the fixture from the loading station to the drilling station and *vice versa*, is effected by opening two doors, one on either side of the fixture. Clamping of the components is controlled by two levers over the fixture. These levers also lock and unlock the two doors. The drill heads are automatically fed and stopped. They work in a set cycle initiated by an electrical push button. The necessity of using two hands for both the indexing and the clamping is an important safety feature. A further operator safeguard is that two doors cover both the cutting tools and the workpieces during the drilling cycles.

Fig. 2 shows the outer stations of the fixture with the components in position and clamped. When loaded on the fixture block, the pistons are located on the bore and supported by a pair of "fingers" locating on the gudgeon pin hole. The components are securely clamped by a vertical bar that fits over the crown of each piston. When unclamped, the bar moves clear of the components and pivots through 90 deg into a horizontal position to give adequate clearance for unloading and reloading. The two-station fixture takes two components at each station, and is mounted on the main casting between the drill heads.

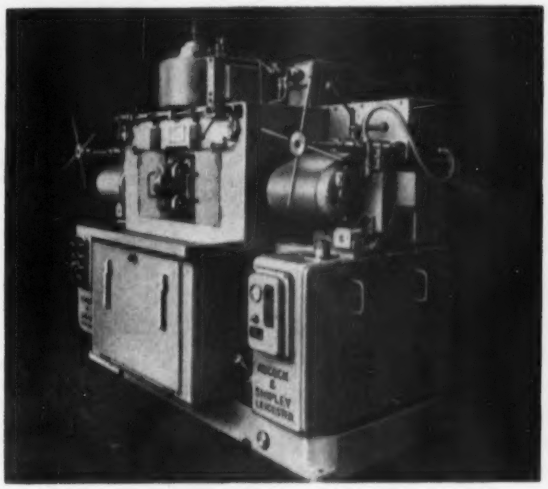


Fig. 1. Adcock and Shipley machine for drilling gudgeon pin holes in two pistons simultaneously

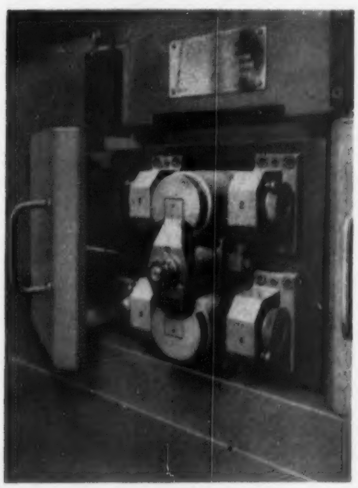


Fig. 2. The loading position of the two-station indexing fixture

The operation sequence is:—

(1) Unload and load two components at the front station.
(2) The two levers at the left and right above the fixture are operated to:
   (a) clamp the components in the front station
   (b) unlock the two doors.
(3) The doors are opened in sequence, first the right hand and then the left. This causes the fixture to index through 180 deg to bring the components to the drilling station.
(4) The doors are reclosed and the two levers are operated to:
   (a) lock the doors
   (b) unlock the clamp at the front station.
(5) Start the drilling cycle by means of the push button.
(6) Unload and reload as in (1).

Once the initial cycle is completed, all further operations follow the same pattern and two completed components are produced each time the fixture is indexed back to the front station. Therefore it will be seen that the total production time for one piston is effectively only half the time of one drilling operation plus one indexing operation.

The machine is fed hydraulically by a servo mechanism, which is operated from cams rotated by a separate feed motor. To allow cycle times to be changed, the feed motor drive incorporates pick-off gears. The two-spindle drill heads are mounted on slides on the cabinet base. Each head is individually driven by a 2 h.p., 1,430 r.p.m. rotor stator unit. Reamers, 0·854 in diameter, are fitted to standard collets which have 1 in individual adjustment. By means of capstan-type wheels, the drill heads themselves have 6½ in manual adjustment.

Drilling is effected in a single stroke of the two-spindle heads. The cycle is: advance at a fast traverse rate of 300 in per minute, then change to a drill feed rate of 0·008 in per spindle revolution at 820 r.p.m., followed by rapid retraction. Provision is made for hand feeding, indexing and clamping for testing and setting-up purposes.

The base of the machine acts as a coolant reservoir. A box section containing the hydraulic control mechanism is mounted on the base. On the extreme left and right of this are two cabinets extending forward. The left-hand cabinet, fronted by its switch panel, houses the electrical control gear, while the right hand cabinet houses the hydraulic power unit. Lubrication is by a one-shot system to the drill head slides. It is operated by a foot pump near floor level at the front of the machine.

From a large capacity centrifugal pump mounted in the base of the machine, coolant is fed to the guide bushes to ensure that powerful jets are correctly directed to the cutting tools. Special attention has been paid to the question of swarf disposal. For this purpose the front of the machine has a large sheet metal cabinet in which is housed a specially designed swarf barrow in which coolant and swarf are collected. The bottom of the swarf barrow is fitted with grids to allow the coolant to drain out. The front of the cabinet is hinged at the bottom to allow the swarf barrow to be pulled out on its wheels for easy removal by fork lift truck.

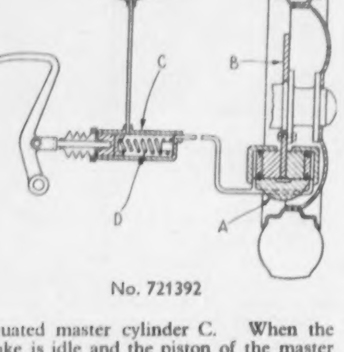
CURRENT PATENTS

A Review of Recent Automobile Specifications

Self-adjusting disc brake

AUTOMATIC adjustment of clearance between friction pads and braking member is provided by this system, without necessitating the use of retraction springs, ratchets, or other devices. The method is particularly intended for disc brakes but may also be used on drum brakes of the external-shoe type.

The disc brake shown in the example is of known type in which the disc is engaged by pads of friction material actuated hydraulically by piston and cylinder mechanisms housed in a yoke A straddling the periphery of the disc B. Operation is by means of a pedal-



No. 721392

actuated master cylinder C. When the brake is idle and the piston of the master cylinder is retracted by the helical spring D, the cylinder is in communication with a fluid reservoir E, which is vented to atmosphere. The internal friction of the system, normally constituted by the sealing rings on the pistons of the wheel cylinders, is so arranged that the hydraulic head is insufficient to move the pistons.

When the system is pressurized by operation of the control pedal, the internal friction is overcome and the friction pads applied to the disc. Heat generated by the engagement causes the disc to expand and the pads and the pistons are forced back very slightly against the fluid pressure. On removal of pedal pressure, the system is unpressurized and the pistons and pads remain substantially in the same positions, although in practice they may be moved back very slightly by high spots on the disc. As the disc cools and contracts it provides, or increases, the running clearance between the pads and the disc. *Patent No.* 721392. *Dunlop Rubber Co. Ltd.*

Torsion-rod suspension

A DIFFICULTY associated with torsion-rod springing is the method of anchoring one end of the rod to the vehicle framing. Commonly the rod end is upset and relatively expensive splining or profiling is employed. The object of the invention is to remove the need for such measures by a simplification of design. A pair of road wheels on opposite sides of the vehicle are sprung by two torsion-rods formed by a single rod bent to a U-shape or V-shape in plan. The loop may be mounted in a transversely arranged central bearing to give a relatively stiffer individual springing, or may be unsupported to provide a stabilizing action between the two road wheels.

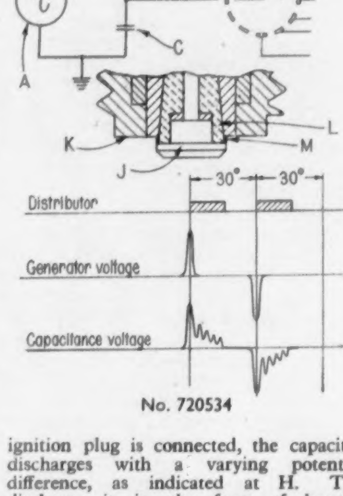
For front wheels suspended on superimposed wishbone links, the spindle at the pivotal axis of one of the links is coupled at A to one end of the common torsion rod B which is mounted in a single, transverse, rubber-bushed bearing C. In an alternative arrangement the torsion rod is extended to take a bearing in the pivot bushes of the wishbone link and a bent-out portion D of the rod engages a cross-bar on the link.

Rear wheels mounted on swinging half-axles are sprung by a torsion rod mounted on each side in longitudinally arranged bearings E. The cranked portion F carries an anti-friction roller G engaging the half-axle casing. With an appropriate geometrical layout, this arrangement can provide a progressive spring rate. For a swinging half-axle having a radius arm H pivoted to the framing, a torsion rod mounted in bearings J has a laterally bent arm K engaged over arm H. *Patent No.* 719993. *Daimler-Benz A. G. (Germany).*

Ignition system

AN engine-driven generator A is arranged to give on open circuit a peaked output voltage of 2 to 3 kV, as indicated by the trace B of a peaked A.C. voltage. A capacitor C ($0{\cdot}05\mu F$) is connected across the output terminals and a high resistance D (33,000 ohms) is inserted between one plate of the capacitor and one side of the generator. The other plate is earthed. Ignition plugs E are connected in turn to the capacitor by an engine-driven, jump-spark distributor F.

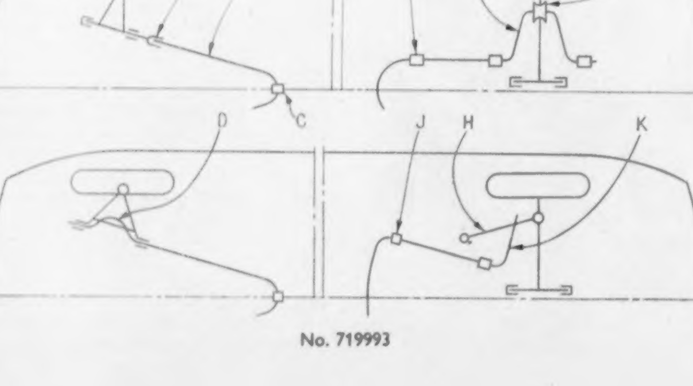
Driven in timed relationship to the distributor, the generator produces one peaked voltage impulse each 30 deg of revolution of the distributor for a 12-cylinder system, as illustrated. The contacts and the contact arm are so arranged that the distributor is in the "make" condition for about 16 deg of the distributor movement, as shown at G. When the



No. 720534

ignition plug is connected, the capacitor discharges with a varying potential difference, as indicated at H. The discharge is in the form of heavily damped oscillations and a long "break" condition is essential to ensure the capacitor is fully discharged.

Ignition plugs have a central electrode J and an annular electrode K with the space between filled with an insulating material L similar to that used in conventional spark plugs. The exposed annular surface M of the insulator, between the two electrodes, is treated so as to be semi-conducting and to have a negative temperature coefficient of resistance. When the capacitor discharges, the distribution of current through the semi-conducting layer becomes unstable, and a large proportion of the current passes through a small section of the layer. This section immediately heats up and its resistance progressively falls until a sparking condition occurs. Such plugs are simple and robust and are immune from the effects of carbon and oil deposition. *Patent No.* 720534. *Rolls-Royce Ltd.*



No. 719993

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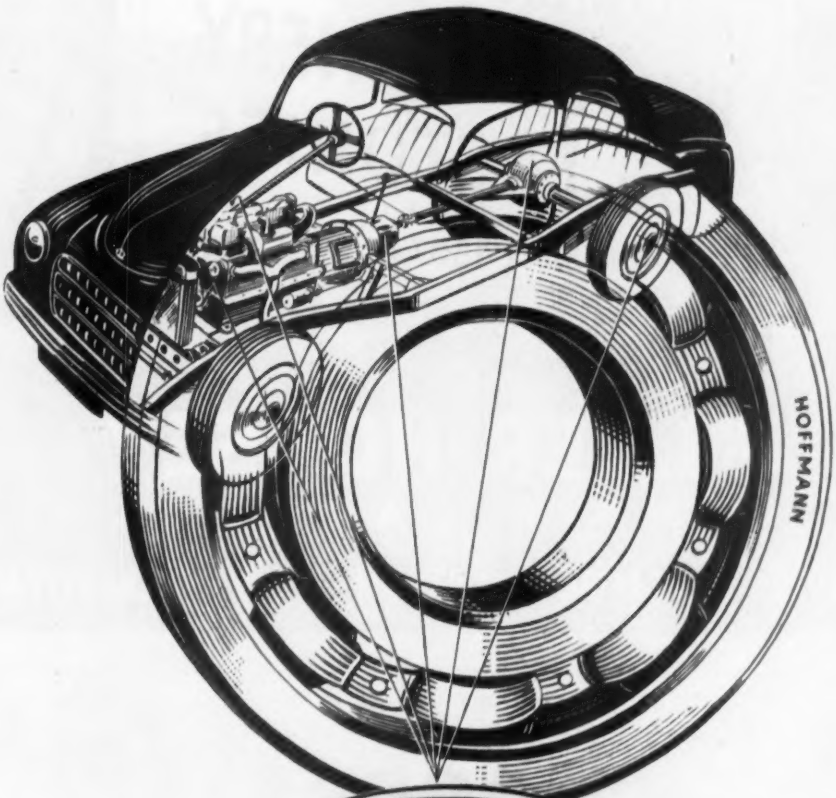

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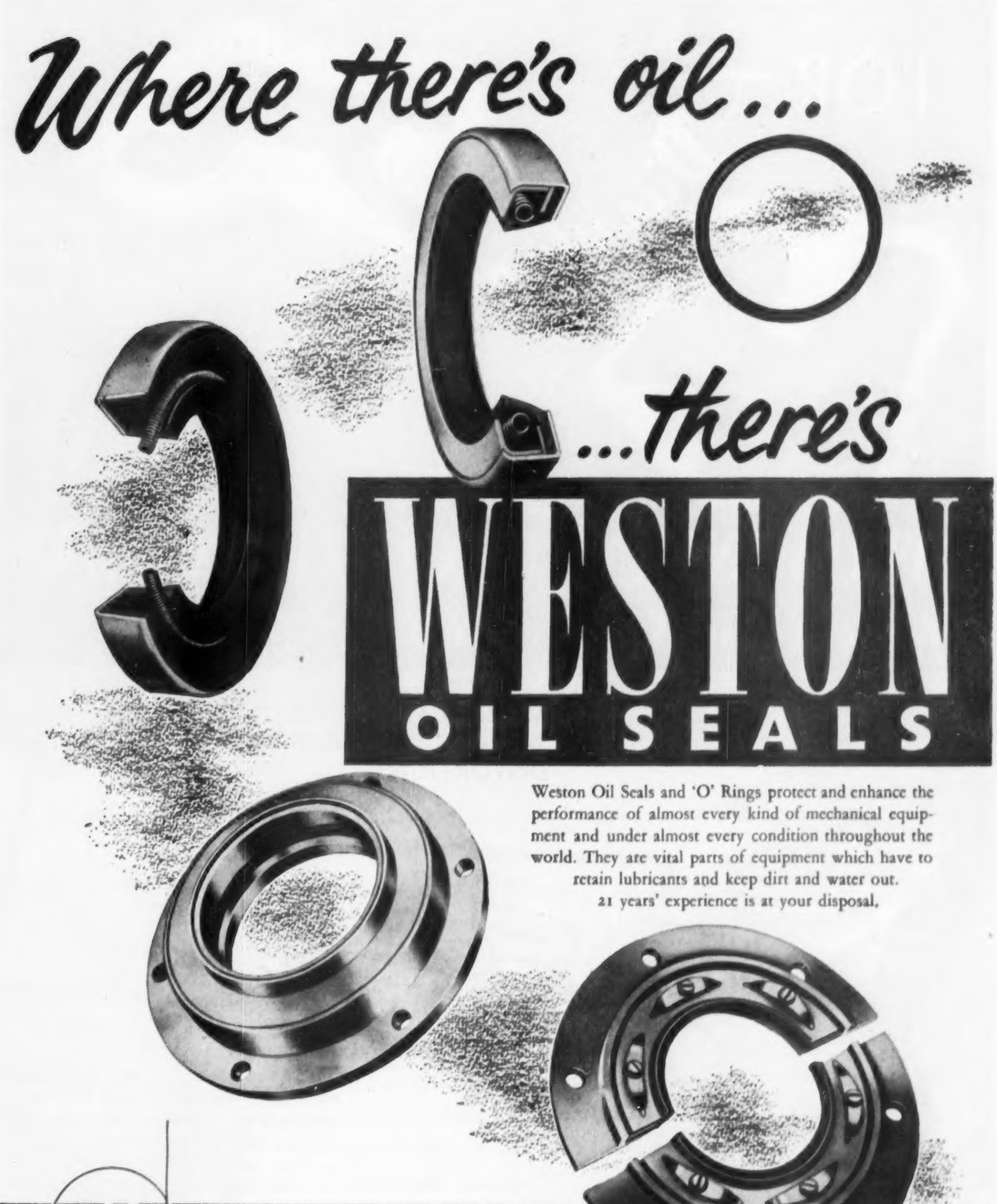
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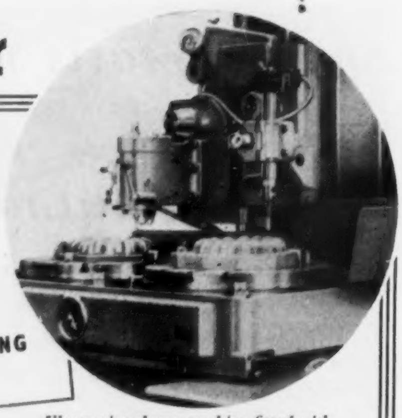

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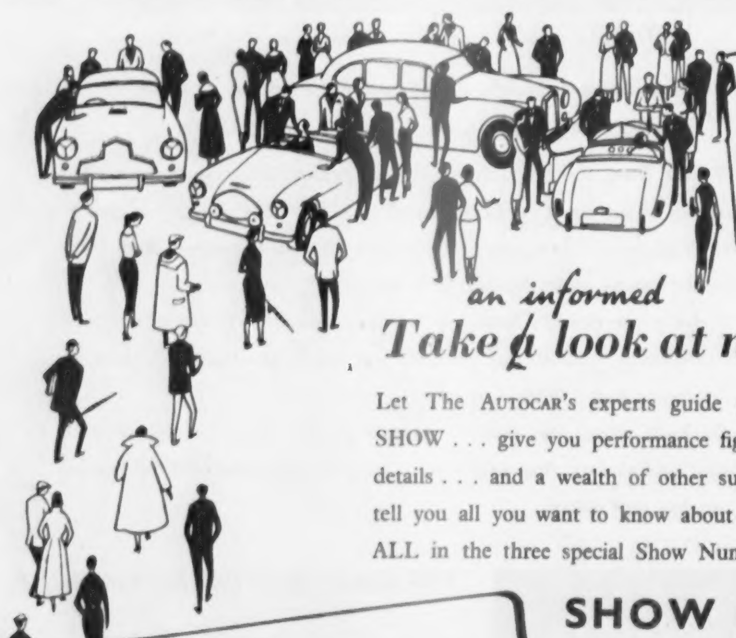

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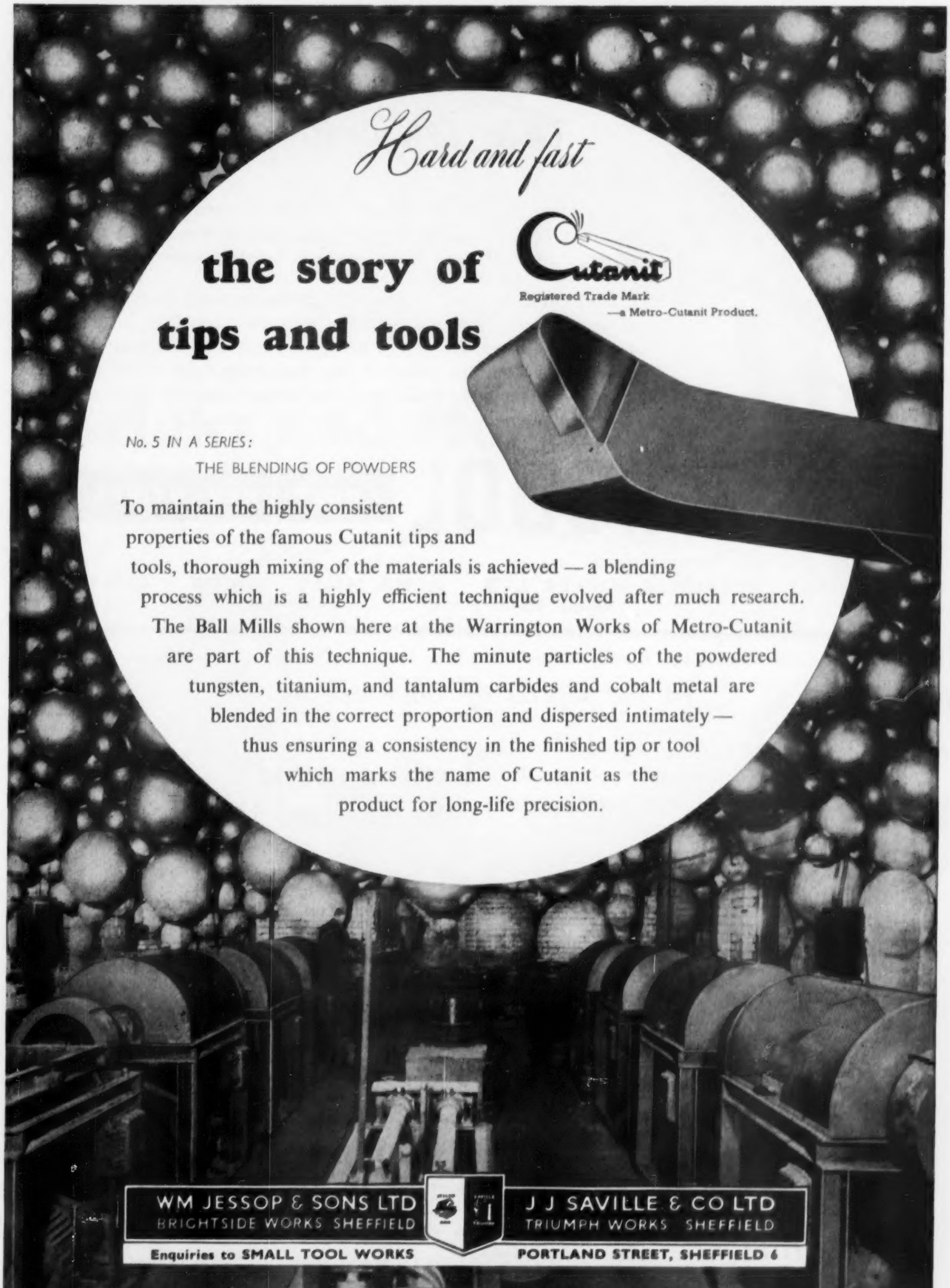
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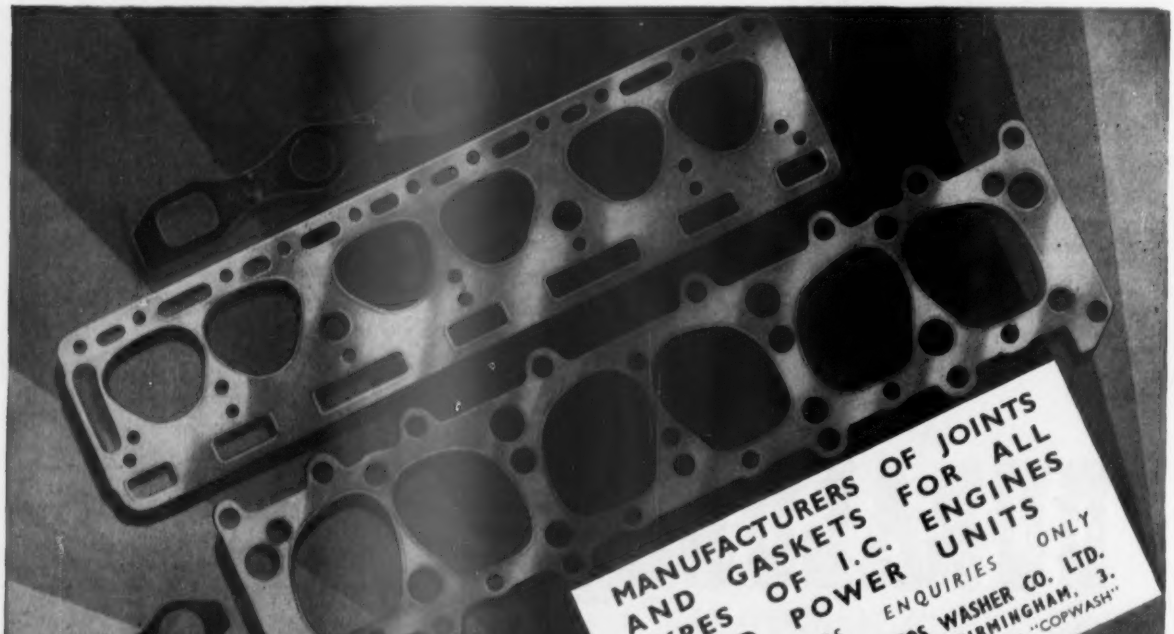

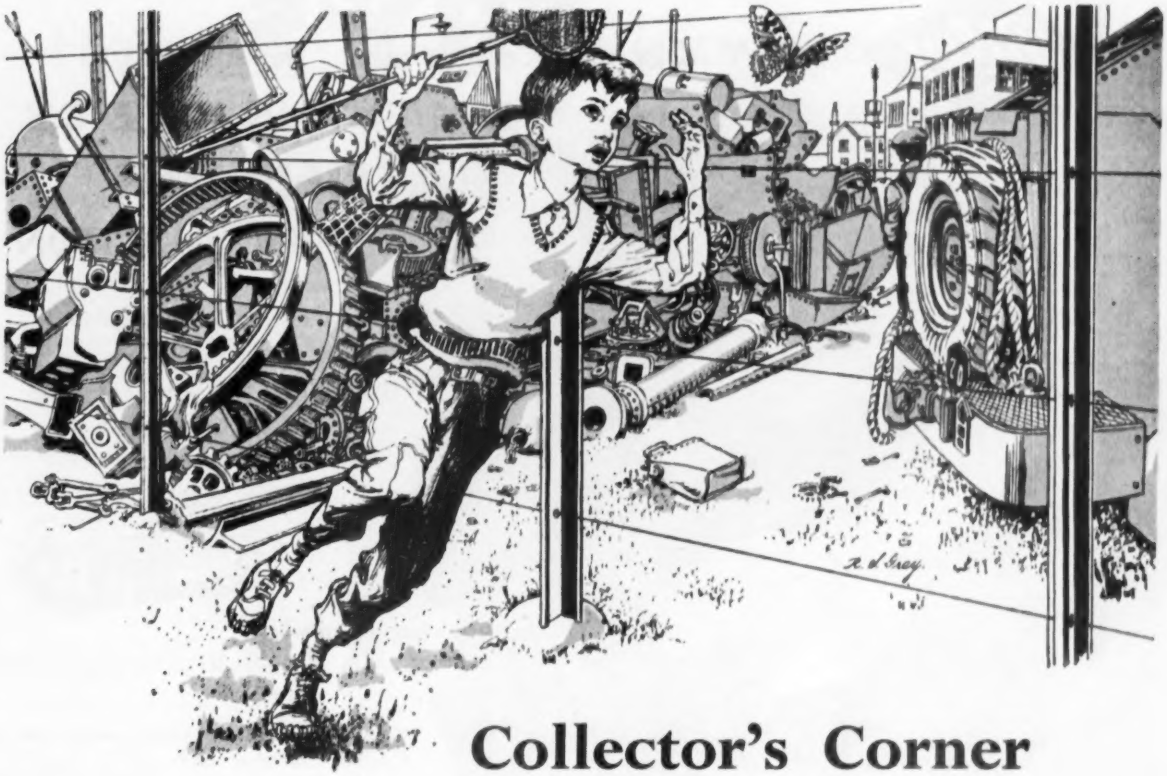

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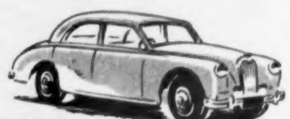

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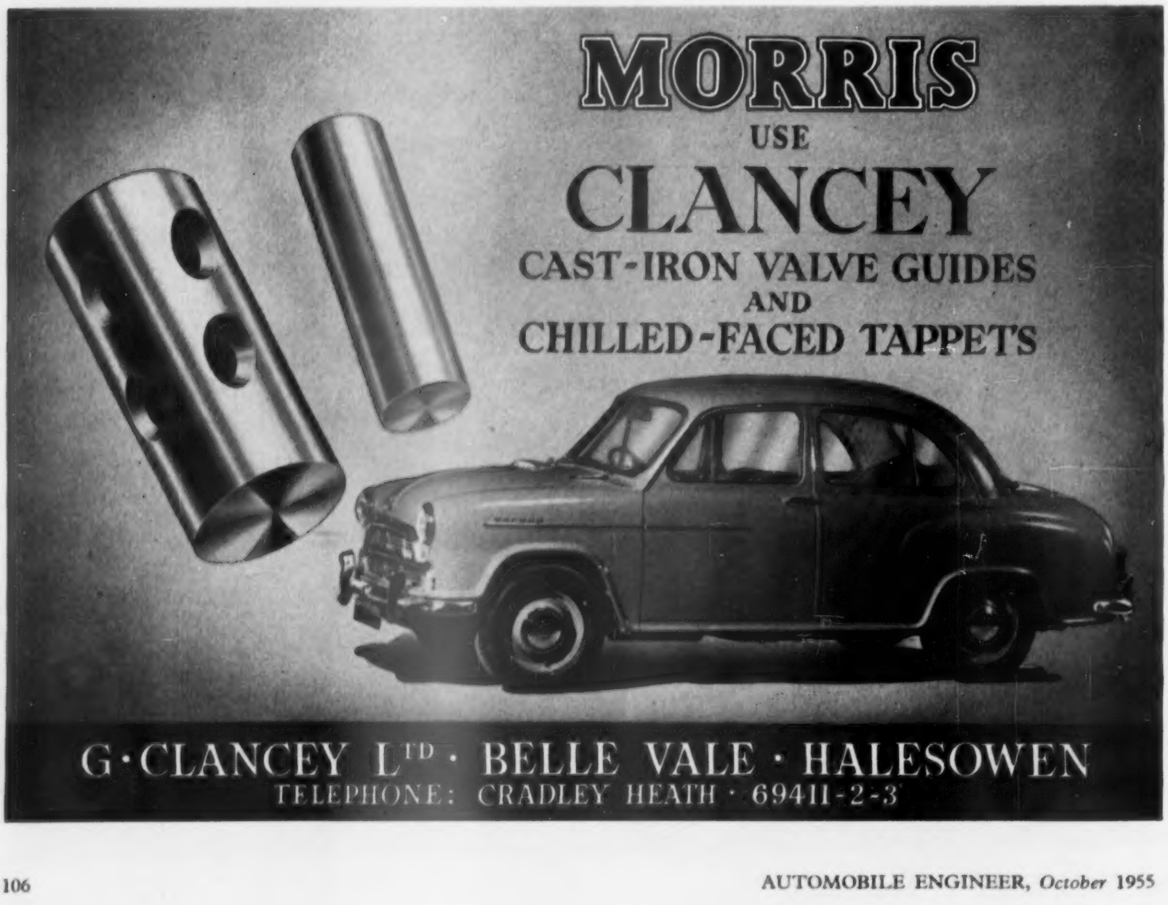

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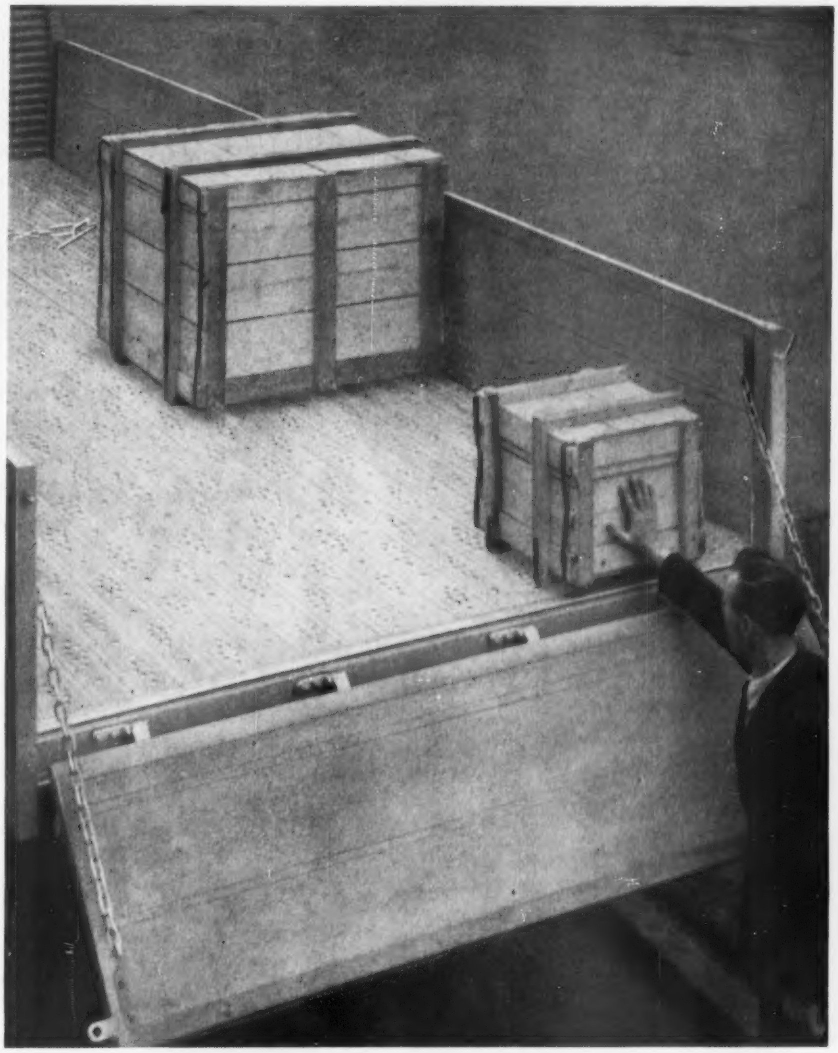

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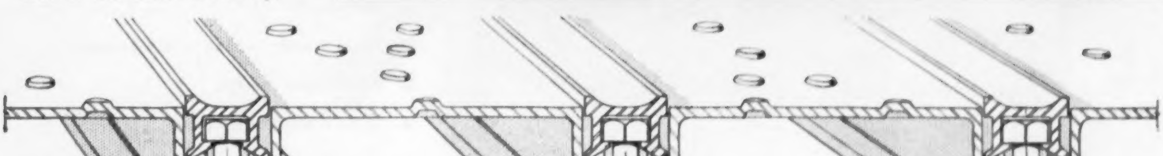

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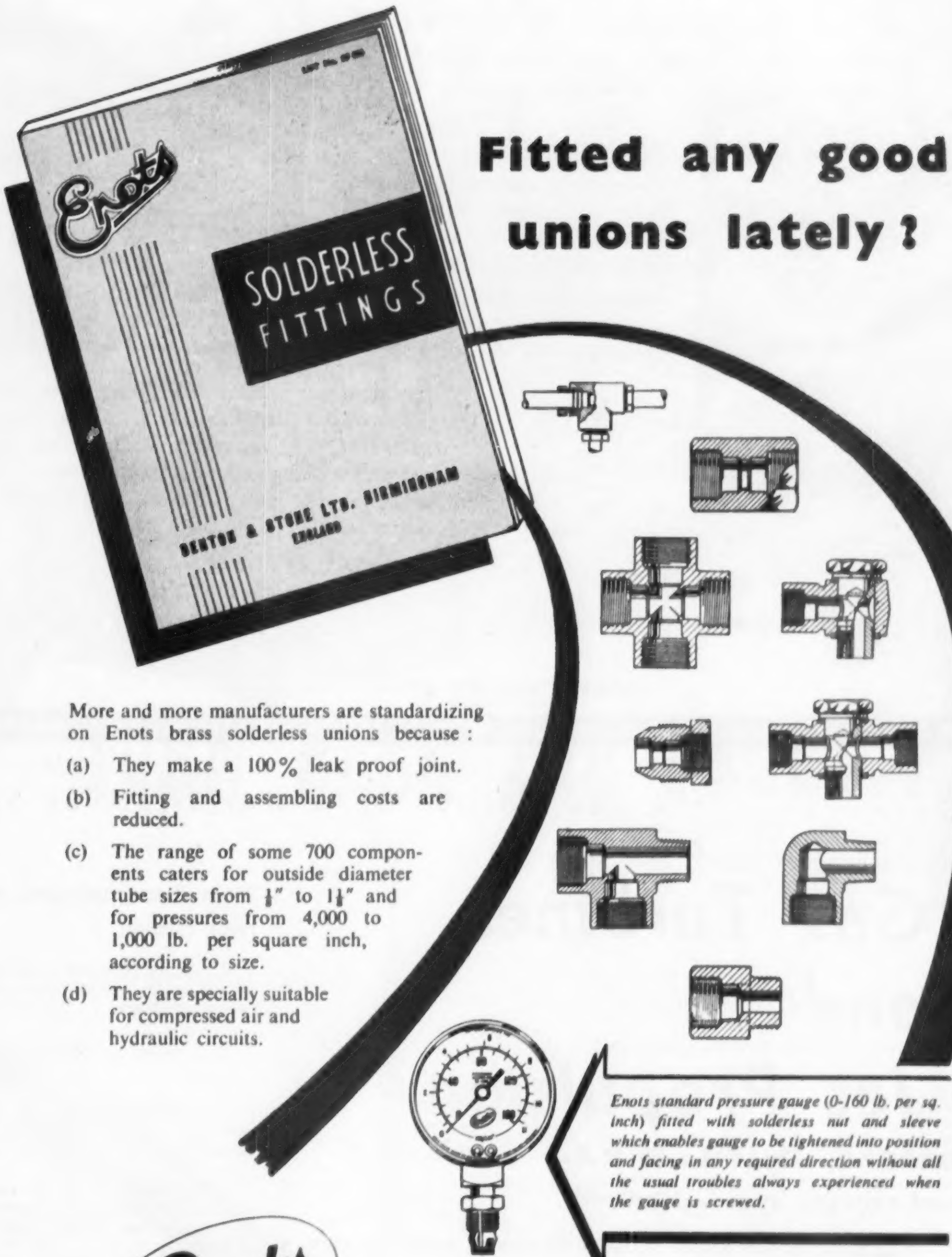

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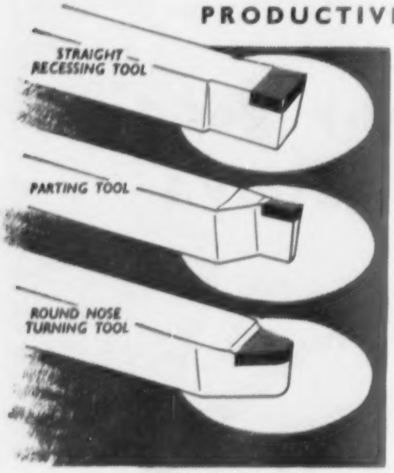

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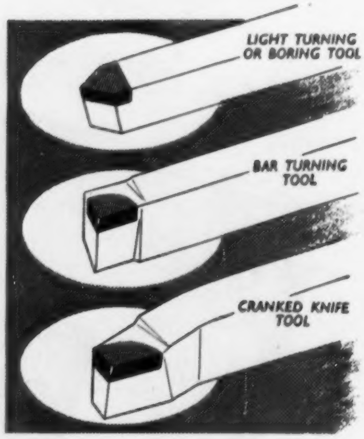

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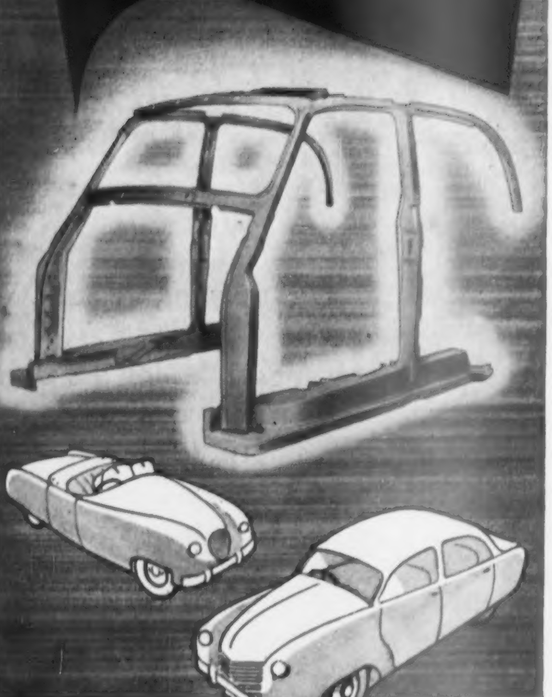

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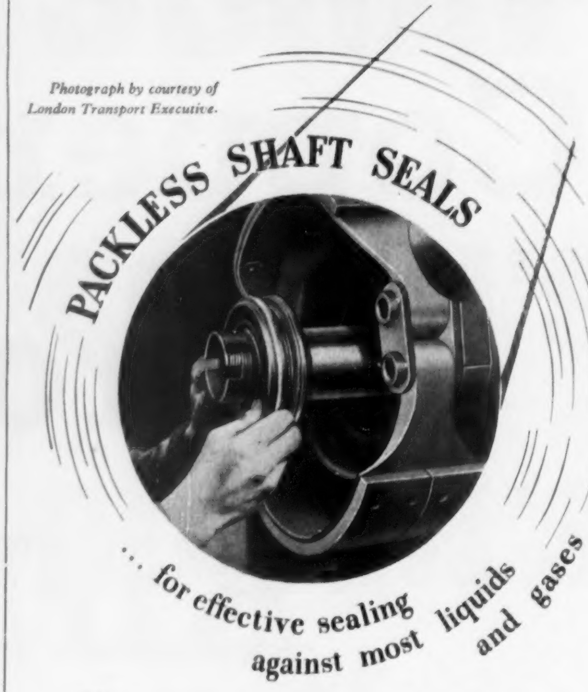

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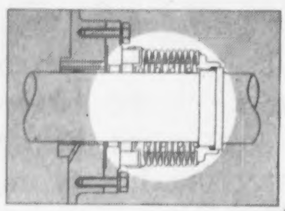

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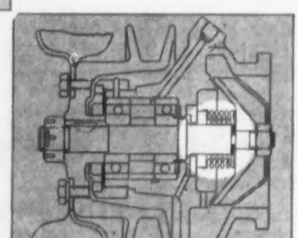

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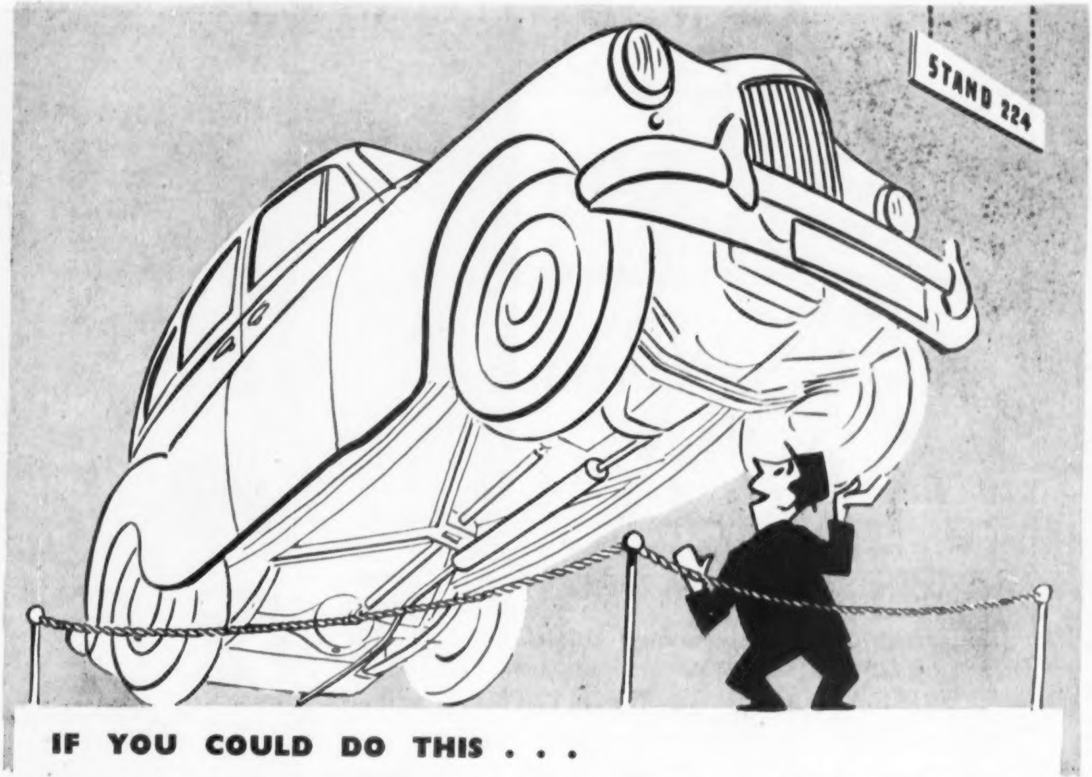

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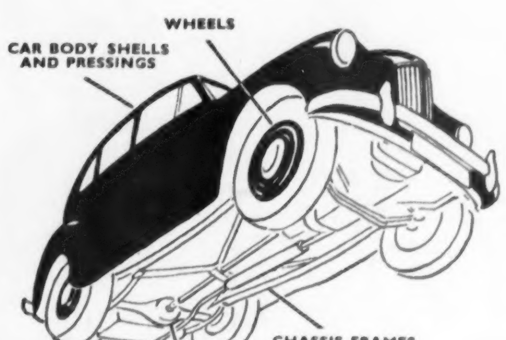

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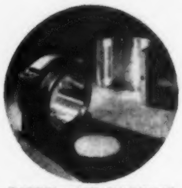

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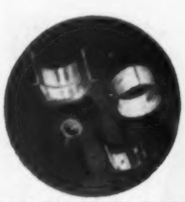

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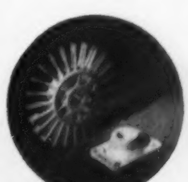

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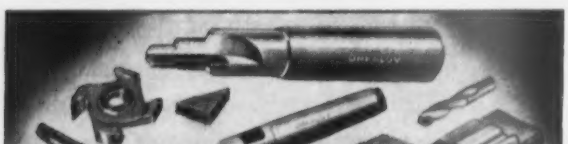
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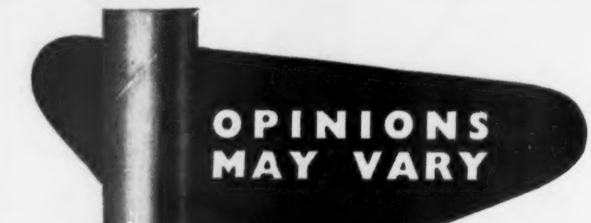
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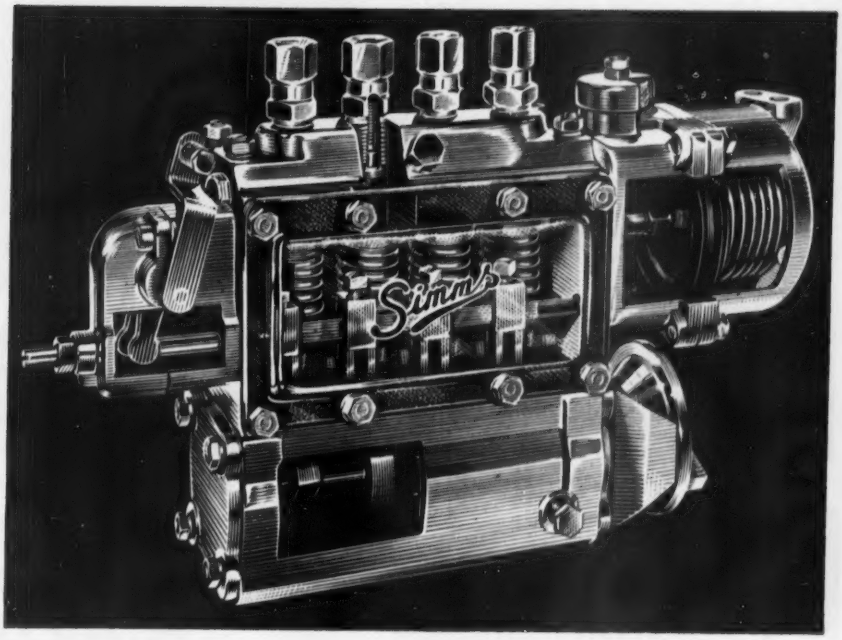

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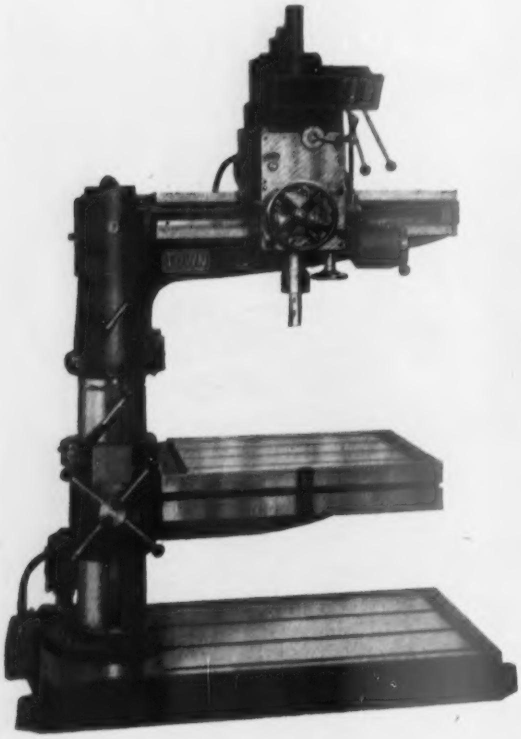

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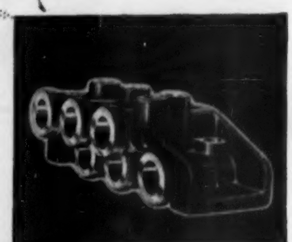

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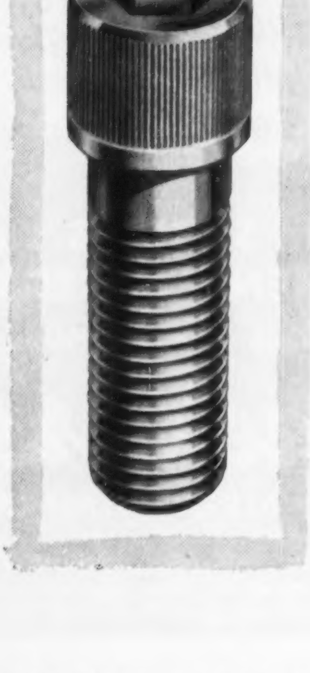

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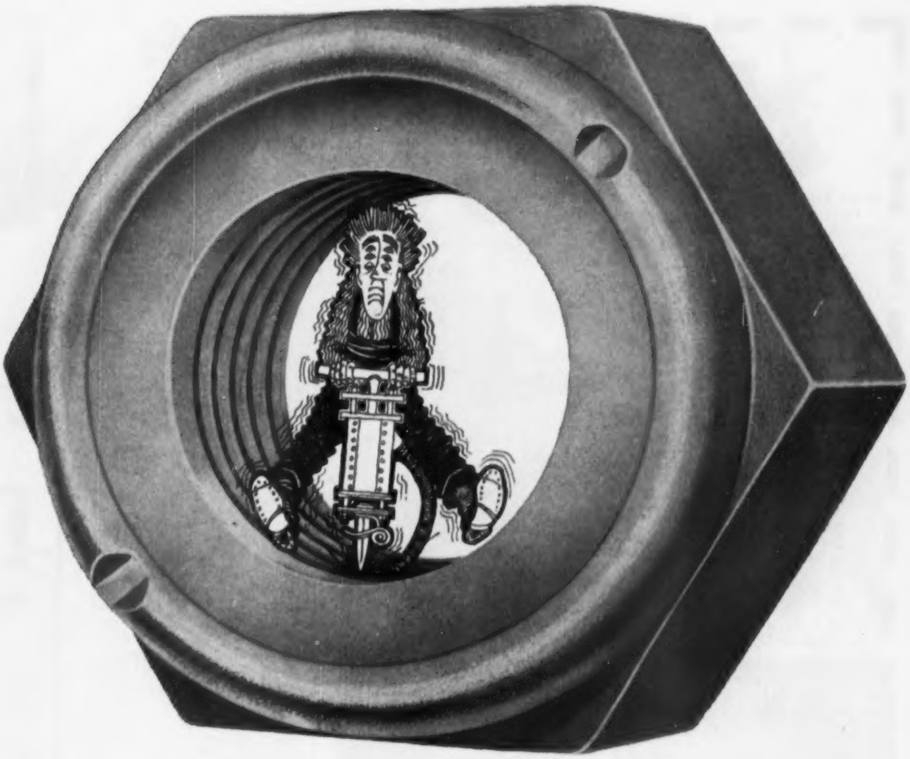

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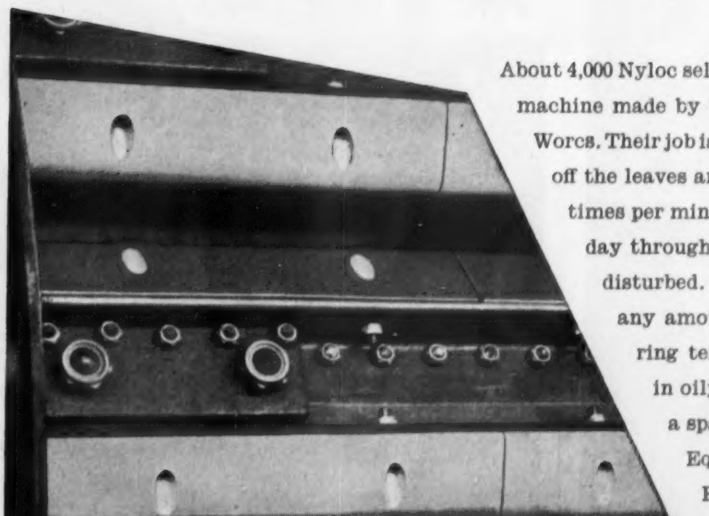

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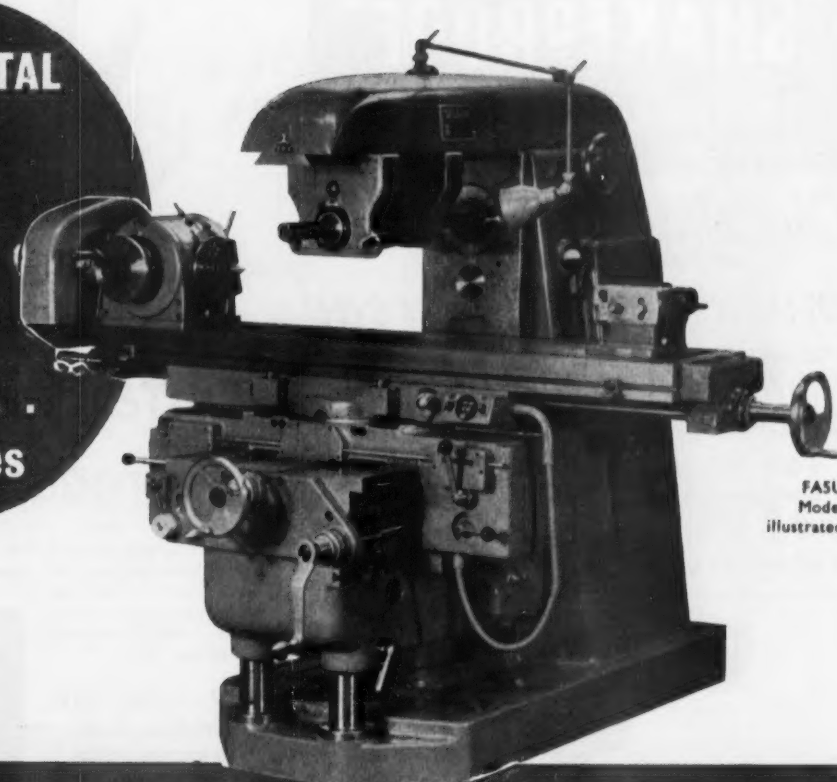

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The engagement of persons answering these advertisements must be made through the local office of the Ministry of Labour and National Service, etc., if the applicant is a man aged 18-64 or a woman aged 18-59 inclusive, unless he or she or the employer is excepted from the provisions of The Notification of Vacancies Order, 1952

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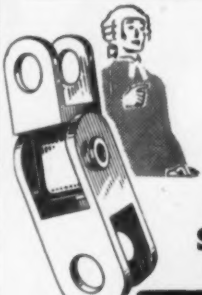

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PLANNING ENGINEER for BUYING DEPARTMENT
Applicants should have served an apprenticeship in Engineering and should, if possible, have passed the Production Engineering Graduates or National Engineering Examination or equivalent.

Previous experience should include mass production shop experience in machine presswork, foundry and forge, with normal methods of planning and costing.

The work will require accurate costing of components used in the Automobile Industry. Age 30 to 45.

ENGINEER for WORKS ENGINEER STAFF
A Machine Tool/Switch Gear Engineer is required urgently.

He will be required to design electrical equipment for machine tools.

PLANNING ENGINEER for PRODUCTION PLANNING DEPARTMENT
Applicants should have had at least four years Engineering and Planning experience and be in possession of H.N.C. in Mechanical or Production Engineering.

Previous experience should, if possible, include some Jig and Tool drawing experience and knowledge of dealing with a large quantity of engineering production. Age 25 to 40.

ENGINEERS for RESEARCH AND DEVELOPMENT DEPARTMENT
Applicants must be in possession of an Honours Degree and previous research experience would be an advantage.

Vacancies exist both in the Gas Turbine and Piston Engine Departments.

SENIOR DESIGNER for TURBINE WORK
Applicants must be in possession of H.N.C. or a Degree and previous experience in Gas Turbine work would be an advantage.

The above posts are permanent with good prospects. Salaries are dependent on qualifications and previous experience.

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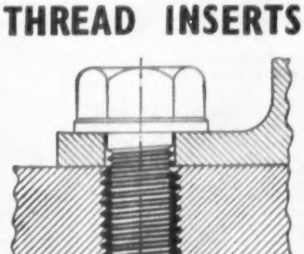

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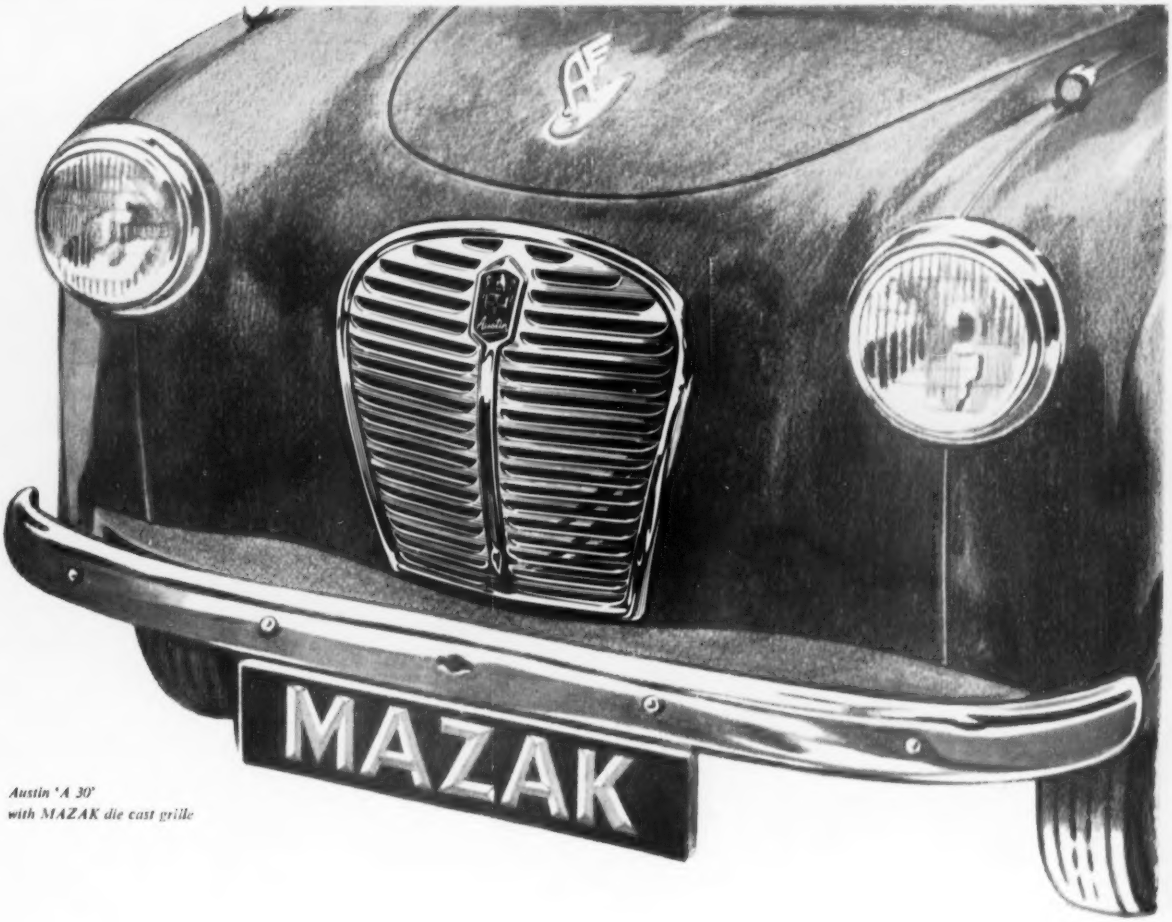

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